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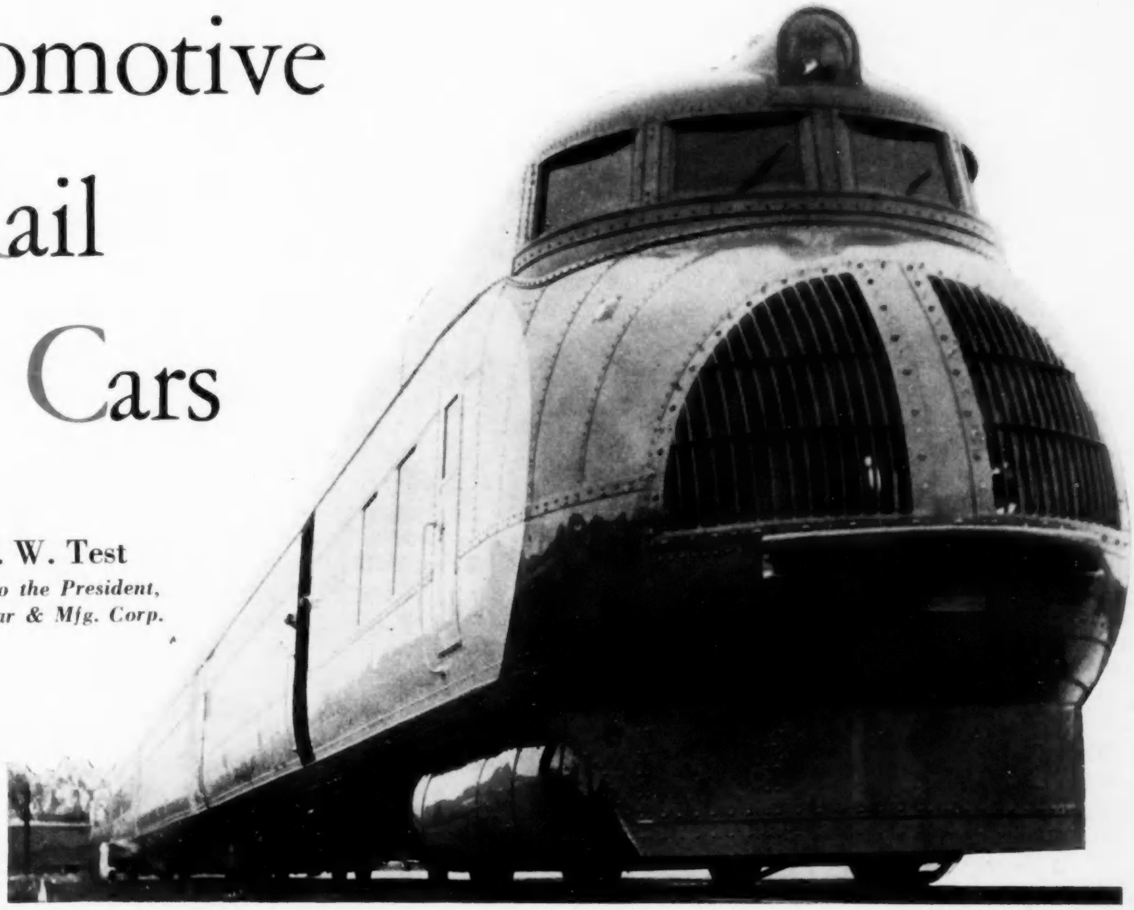
Automotive Rail Cars



Wide World Photos

By E. W. Test
Assistant to the President,
Pullman Car & Mfg. Corp.

COMPETING modes of transportation did not begin to be felt by the railroads until the advent of hard-surfaced highways about ten years ago. During these last ten years railroad passenger business has been declining steadily. There has been an average decline of 1000 railroad passenger miles per year for each additional automobile registered since 1920.

A first-class train, including Pullman sleepers and observation car, weighs about 2,000,000 lb. and carries an average load of 100 passengers. A 20,000 lb. bus will carry about 30 passengers. An automobile weighing 4000 lb. will carry five passengers. These figures show that a steel passenger-train requires from 25 to 30 times as much weight per passenger at its competitors.

With this great disparity between weights-per-passenger the railways are falling behind competing modes of transportation in the element of speed. It is impracticable to increase locomotive power with the present standard type of equipment to keep up with the increased demand for speed without an inordinate increase in operating cost.

An obvious solution appears to be development of light-weight high-speed streamlined trains which can be operated at a lower expense. The Union Pacific Railroad and the Pullman Car & Manufacturing Corp. have undertaken such development. The three-car train of this type recently completed at Pullman consists of a forward car containing a 600-hp. distillate electric powerplant, a 33-ft. mail compartment and a small baggage room, of a second car which is a coach, seating 60 passengers; and of a third car which is also a coach with a buffet in the rear for serving light meals to passengers at their seats. Seats are provided for 56 passengers.

A study was made of all available materials, including aluminum alloys, stainless steel, and other steel alloys with physical properties intermediate between ordinary steel and stainless steel. In order to obtain extremely light weight, the choice narrowed down to aluminum alloys or stainless steel. It was finally decided to use aluminum alloys for the entire car structure, except for the bolsters, articulation castings and truck frames, for which purpose there was used a special-alloy cast steel having high tensile-strength, high yield-point, and great ductility. The net result is a three-car train weighing 165,000 lb.

Aluminum plates can be formed readily for the curved surfaces used in connection with streamlining, and can be riveted readily and spot welded without injury to the material. All of these factors contributed to the selection of aluminum for this first development though, since that choice was made, several new steel alloys have been developed by the steel manufacturers.

In the study of streamlining, advantage was taken of work done along these lines in connection with aircraft development. It was soon learned however that the streamlining of the underside of the train was an all-important factor, as eddies created between the irregularities of the road bed and the underside of the car introduced considerable drag on the powerplant. It was therefore necessary to set up modifica-

[This paper was presented at a meeting of the Syracuse Section, Feb. 19. In the absence of Mr. Test it was read by Hubert Walker, chief engineer, American-LaFrance & Foamite Corp.]

tions for tests in the wind-tunnel in order that these new factors could be investigated properly.

Wooden models of the train were built with detachable fronts and rears having various shapes and these were all subjected to wind-tunnel tests. The final form for the train was determined from the results obtained in these wind-tunnel tests. The power required for the speed desired also was determined fairly accurately.

Streamlining Offers Difficulties

The streamline models indicated a smooth canopy and closing up the gap between sections, which is, of course, absolutely essential; but the accomplishment of this was a difficult task in designing the actual train, taking into account the relative angularity between cars on curves. An aluminum shield, which is the prolongation of the car contour, extends from the rear end of the forward section toward the front end of the following one. The extent of this projection is contingent on the minimum radius of curve to be negotiated. Closing up this gap between the hood projection and the following car section is a rubber sheet rigidly attached to the following car section, which assumes the contour of the car and is free to move at its forward edge. Spring-actuated arms mounted on the drum portion of the articulation, with rollers bearing on the inner side of the projecting hood, keep this rubber stretched to close up the gap between the hood and the following car section.

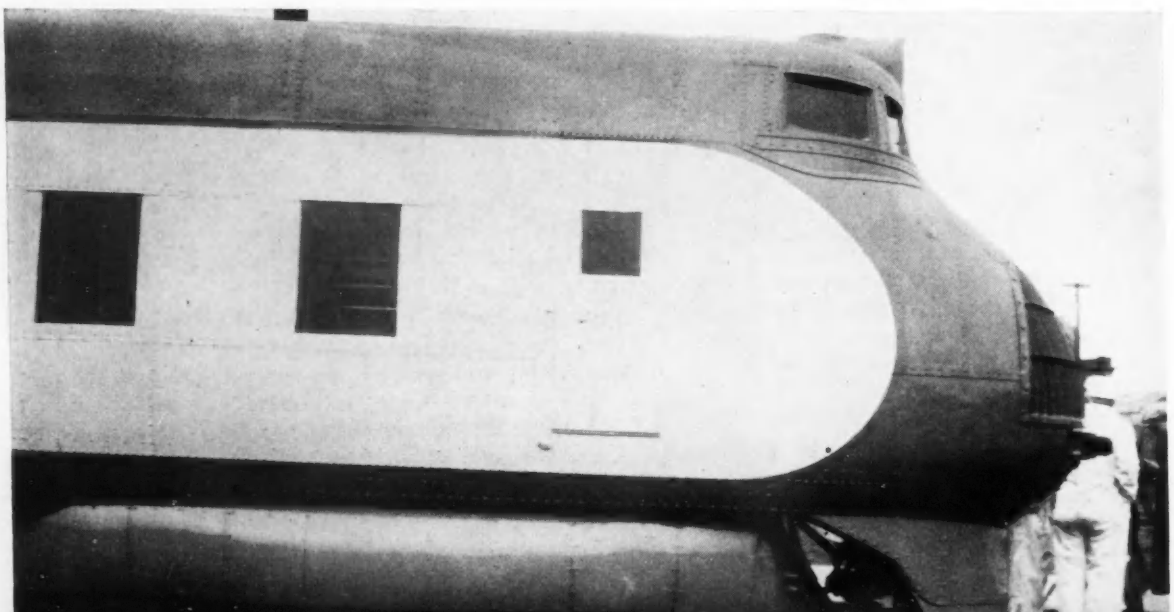
To procure the greatest strength with the least amount of material, the car cross-section selected was as near to a tubular design as possible—consistent with streamlining for cross winds. The outer surfaces are of aluminum-alloy sheets and the framework of extruded aluminum-alloy sections. The entire car-body structure is designed as a unit to withstand either draft or buffing stresses. A very large moment of inertia is obtained with this near-tubular cross-section. Deflections are thereby very closely controlled. The longitudinal members extend the entire length of the car and the transverse members are in one piece for at least one-half of the cross-section. In addition to the primary car structure, the duct at the ceiling used for the distribution of air for air conditioning is also designed to act as a compression member. Along the center of the car underneath the floor is a built-up structural I-beam section acting as a tension member in coordination with the upper duct structure which is in compression.

The hazard of grade crossing collisions with automobiles, trucks, etc., has been carefully considered in the design of the front end. Approximately one-half of the total weight of the train or 80,000 lb. is carried on the front trucks. This is more than is carried on the front truck of a passenger locomotive and necessarily requires massive supports for the engine at the floor line. These massive structural members extend to form a strong parabolic arch designed to resist damage from highway crossing collisions.

To obtain further weight reduction and smooth riding, articulation between cars has been resorted to. This makes for less weight and frictional train resistance and also avoids the necessity for the use of couplers, draft gears, and a complicated vestibule arrangement. It also retards the independent oscillation of each car.

Articulation is effected by attaching an extension casting to each adjoining end sill. These castings terminate in center plates which rest one on top of the other and they in turn rest on the truck center plate. A heavy locking king pin secures all three plates together. The center plates are lined with Oilite bronze. Special design of side bearings employing rubber in shear are used to deaden oscillation and promote smoother riding.

The trucks are all of the four-wheel type. Wheels on the front truck are 36-in. diameter rolled steel. The remaining trucks are equipped with 33-in. rolled steel wheels. Roller-bearing journals are provided throughout, being placed outside of the wheels on the front truck due to space necessary for the two 300-hp. ventilated motors geared to each axle. The roller-bearing journals on all other trucks are placed inside of the wheels to reduce air resistance. The armature shafts and the driving motors are mounted on roller bearings. The



Power car of the Union Pacific three-car train. Light areas are painted yellow, with tan contrast. Note shrouding of wheel-trucks for reduction of wind resistance.

armatures are wound for a safe maximum speed of 110 m.p.h.

As wind-tunnel experiments developed the fact that shrouding of the trucks would reduce wind resistance about 20 per cent, such shrouding has been adopted. By the use of deflectometers and extensometers the engineers have studied truck action to obtain a basis for correct design. Localization of stress is especially avoided in such light-weight structures and all material is used only where it can serve a useful purpose. Special consideration is given to impact stress as, in many instances, such stresses occur where they are least likely to be expected.

The trucks are of cast steel, the frame and transom being cast in one piece. An alloy of extreme ductility was used, having a minimum yield point of 50,000 lb. per sq. in.

Trucks of this train are designed to incorporate the use of rubber to its fullest extent where there is sufficient background to warrant its adoption. There is no metallic contact between the frame and any part of the journal box. The sides of the journal box are extended either side of the axle and between these extensions on the side frame are rubber "doughnuts" under compression, which have ample capacity in shear to perform their part in supporting the load acting upon the truck, thereby cushioning all rail shocks. Coil springs work in parallel with the rubber so that ordinarily the rubber "doughnuts" are lightly stressed. The "doughnuts" are useful in absorbing both vertical impact and fore-and-aft forces due to stopping and starting. The pedestal coil springs are mounted on rubber pads to kill metallic contact. Traction and braking forces are further cushioned by the use of rubber in combination with the chafing plates between the bolster and transom. The use of rubber in the wheels has been avoided as it is felt that designs so far are in the experimental stage.

Braking Carefully Studied

Careful study has been given to the matter of braking. At speeds of 100 m.p.h. effective braking becomes imperative not only for obvious reasons but also to avoid the necessity for change in the arrangements of signalling systems. Heretofore it has not been possible to obtain uniform maximum retardation as the amount of braking effect depended upon the skill of the locomotive engineer in making the proper original application and subsequent reductions. It is well known that the coefficient of friction between brake shoe and wheel varies with speed through a very wide range, this coefficient decreasing rapidly at the higher speeds. The effect is greatly augmented at still higher speeds. In order to provide a uniform rate of retardation and to maintain it at a safe maximum it is necessary to have control of brake-shoe pressure in proportion to the speed. The new brakes provide this by controlling cylinder pressure automatically with a simple device known as the decelerometer. This is a simple but effective device only recently developed although it has been thoroughly tested in service. This instrument consists of a movable weight of about 100 lb. sensitively mounted on ball-bearing rollers arranged to move in the line of motion of the train and suitably restrained by a calibrated spring. This weight acting through levers and valves controls the brake cylinder-pressure in proportion to its inertia and therefore in proportion to the brake retarding effect on the train.

It is possible to obtain a constant retarding effect from the maximum speed of the train until it comes to rest. However, this would not make for passenger comfort as a noticeable jolt would be felt at the end of the stop. In order to eliminate this the rate of deceleration is gradually changed to a low value just previous to the stopping of the train. This is all accomplished automatically. By the use of this device trains can be stopped from exceptionally high speeds in shorter distances than those obtained on existing trains running at much lower speeds and it is accomplished without discomfort to the passengers.

The air brake system is of an entirely new design combining electric and pneumatic control in parallel. The pneumatic feature is based on a two-pipe circuit consisting of a supervisory line and a control line. The supervisory line maintains a maximum pressure at all times in the individual reservoirs under each car. In conventional brakes it is not possible to charge the reservoirs during brake applications. The control line is used to apply and release the brakes by admitting air to the electric-pneumatic relay valve under each car, this valve controlling communication between each brake cylinder and its adjacent reservoir and from the cylinder to the atmosphere. This control line passes from the engineer's brake valve through the decelerometer valve to the relay valve. The pneumatic control line is parallel with an electric circuit actuated by contact points on the engineer's brake valve which operates a magnetic-control feature on each electric magnetic relay valve. This electric feature synchronizes and accelerates brake applications and releases. The use of an electric-pneumatic control provides adequate protection against operating failures. This is necessary owing to the use of a straight air-brake system.

Aluminum Brake Cylinders

Brake cylinders are of aluminum and mounted on the trucks, two cylinders per truck. Each cylinder acts on one pair of wheels only. The front or motor truck is equipped with clasp brakes while the trailing trucks employ single brakes. Brake levers and beams are made of aluminum alloy.

Air conditioning and ventilating is provided through a duct on each side below the floor line and a central ceiling duct throughout the entire train, all of these ducts being connected between cars by flexible bellows. When heating, the air is fed to the cars from the floor ducts and exhausted through the ceiling ducts. When cooling, the air motion is reversed being delivered to the car from the ceiling duct. Heat is obtained partly from hot water used for cooling the engine and from an oil-fired hot-air furnace installed in the baggage compartment. The hot-air furnace is capable alone of furnishing sufficient heat for the train.

An electric motor-driven Freon compressor is installed in the baggage room for cooling the air when desired. The air is circulated at all time whether cooling or heating is required and approximately 25 per cent fresh air is introduced into the cars to obtain sufficient air changes and to exclude dust by maintaining a slight pressure in the car. The windows are set nearly flush with the outside of the car surface and are made of safety plate-glass and sealed in rubber. Rokflos 2 in. thick is used for insulating completely around the shell of the car and in the ends. Indirect lighting is provided by means of a trough on each side of the ceiling duct. This trough, together with a properly curved ceiling, distributes the light evenly at all reading positions. Three intensities of light can be obtained, the lowest of which is intended for night lighting while passengers are sleeping. The seats are provided with four reclining positions and the space underneath the seats is kept clear to permit storage of baggage.

Communication between train and engineman is afforded by electric signals. The headlamp is streamlined into the roof of the engineman's cab. It is provided with two beams, one

horizontal and one vertical. A 10-in. reflector directs one beam vertically so that it may be seen from a greater distance than the horizontal beam. In addition to the usual type of siren for use through towns a powerful signal with long audible range is used for high speeds.

The powerplant for this train consists of a distillate-burning engine, developed especially for this service by the Winton Engine Corp. It is of the 12-cyl. V-type, with 7½ in. bore and 8½ in. stroke, rated 600 hp. at 1200 r.p.m. The entire engine frame, including the crankcase and cylinder water jackets, is of welded steel construction. The 5½ in. diameter crankshaft, which is dynamically and statically balanced, is made from chrome-nickel-molybdenum steel, having an elastic limit of 130,000 lb. per sq. in., with a Brinell hardness of 300. There is no cast iron used in the construction of this engine, excepting the cylinder sleeves, which are made of a special grade of cast iron suitable for this purpose.

Special Carburetors Used

The distillate fuel is handled by special type carburetors of the multiple jet, fixed air-ratio type, which have been developed largely on the Union Pacific Railroad over a period of 18 years by the Duff Engineering Co., and are especially adapted to handling this heavy fuel. There is a separate carburetor having 10 jets, attached directly to each cylinder head, and atomization of the fuel is accomplished without the application of heat. The fuel is supplied to the carburetor by electric-driven turbine pumps with gravity return to the fuel tank, and as floats and needle valves are not used, there is no surplus fuel carried in the engine room. Fuel capacity is sufficient for a 1200-mile run.

To further facilitate streamlining, the cooling radiators are located under the ceiling of the engine room, and cooling is effected by having a closed engine room under forced draft. This also supercharges the engine to some slight extent.

The motorman, with all of his controls and necessary instruments, is located in an elevated cab, which is separated from the main engine room. From his elevated position he will have an increased visual range and be removed from the noise of the powerplant. His main operating controls consist of a throttle, a controller and a brake valve. The brake valve is equipped with a "deadman" control, requiring a motorman to keep either a hand or a foot on this control while train is running, and if for any reason hand or foot is removed from this control, there is an immediate automatic closing of the throttle and application of brakes. The direct-connected generator is connected to the front end of the engine through a flexible coupling and is capable of developing approximately 425 kw., and the current, through remote control, is led directly to the two 300 hp. traction motors on the front truck.

In addition to the main generator, there is an engine-driven auxiliary generator having a capacity of 25 kw. at 76 volts, which supplies current for battery charging, one of the two air compressors, air conditioning and lighting. An additional 8 kw. engine-generator set at 76 volts is carried in the baggage compartment for service when the main powerplant is shut down.

The railroads must place themselves in a position to meet competition in passenger traffic. They at present possess the inherent advantages of safety, comfort and reliability. What else must they do to retrieve lost passenger traffic? It appears that they must increase the speed of their trains and economy of operation. This new high-speed service will go a long way toward meeting these requirements.

Progress in Rail Car Design Continues

THE rail car is about as old as the automobile. In the period from about 1907 to 1912, two distinct designs were developed. About 160 to 170 units were built. From 1914 to 1921, War activities brought a stop to the development.

Beginning about 1921, the railroads were much interested in rail cars, originally as a substitute service for use on branch lines where service must be continued by reason of legal requirements, but where actually the volume of business had dropped off to the point that steam trains were unprofitable. The first rail cars developed for this service were of comparatively small capacity and in most cases used units at least some of which were more or less of the automotive type.

These units having proved to be successful within their limited field, the demand was created for constantly increasing sizes, powers and speeds, which would take care of still heavier services, so that a series of cars, each larger, heavier and more powerful, was developed through the period ending about 1929. These cars were almost all used for secondary services, either on branch or secondary lines or for local services on the main lines. The primary object was economy in operation.

Some of the more alert of the railroad men realized that if suitable equipment could be built which could be operated in such a way as to save time for the passenger, transporting at greater safety and comfort, and at the same time at a lower operating cost more or less competitive with bus costs, a new field might be opened up for the rail car, or, perhaps better, motor train. A considerable amount of thought has been given to these possibilities, as evidenced by the number of experimental units that have been built of various types, ranging from extremely light single cars up to and including 9-car Diesel powered trains intended for transcontinental service, which are now on order.

Railroads Have Opportunity

It seems that if the railroads use suitable equipment, operated with reasonable frequency and at sufficient speed, they can well expect to regain a great deal of the passenger travel which has been lost to the private automobile and, to a lesser extent, to the motor bus, on runs of such length that the journey can be made at least as quickly by train.

This would indicate the possibility of considerable revenue on service operating between major cities at a comparatively high schedule-speed, when these cities are at a distance such that the railroads can compete with or improve upon the time made by other means of transportation. Just what the minimum distance may be where such service is justified is probably debatable and undoubtedly would vary with the various local conditions. In operations such as Detroit to Chicago, Chicago to St. Louis, Chicago to Indianapolis and Cincinnati, are examples of the sort of service where this class of equipment might be used to advantage, both for the operator and the passenger.

It would seem that for any services, particularly low cost service above the minimum distances, the motor train can very well find a field.

—**Charles O. Guernsey**
Chief Engineer, J. G. Brill Co.

Abstract of a paper on "Rail Car Development" presented at a Philadelphia Section Meeting, April 18.

Chronicle *and* Comment

By
Norman G. Shidle

REDUCTION of 59 per cent in number of sizes is accomplished by the standard for twist drill sizes from 0.0156 in. to 0.5000 in. published in the December issue of the S.A.E. JOURNAL and unanimously adopted by the S.A.E. Standards Committee as a standard to be published in the S.A.E. HANDBOOK, C. W. Spicer points out in a letter containing interesting comments on this recent Society achievement. Mr. Spicer is chairman of the General Sectional Committee on Small Tools and Machine Tool Elements, under whose auspices this standard was developed.

Before this standard was adopted, Mr. Spicer writes, a questionnaire was mailed to more than 200 of the largest users of twist drills in America. Overwhelming approval of the program came from those replying, thus giving the proposal the practical stamp of user consent.

This standard now is before the American Standards Association for final adoption, already having been approved by the three sponsors—the S.A.E., the American Society of Mechanical Engineers, and the National Machine Tool Builders Association.

POINTING out that "the country is rapidly approaching a limit to the capacity of consumers' goods manufacturers to absorb unemployment," and emphasizing that "further substantial progress in reabsorbing the unemployed necessarily depends upon the revival of capital goods industries," the Consumers' Industries Committee had this to say in its interesting report to General Johnson:

"What then is the trouble in the capital goods industries? . . . It is not sufficient answer merely to say that surplus capacity has existed. The consumers' goods industries themselves would, in our judgment, have substantial demand for new capital in the way of repair, replacement and modernization without adding to productive capacity, provided business could have normal and reasonable confidence. We believe that throughout the greater part of our entire economy there is potential, legitimate demand for new capital which awaits the return of the confidence factor.

". . . What retards and destroys the confidence factor? . . . the repeated and unnecessary stirring up of uncertainties and fears which, while assuming political form as legislative proposals, are pertinent because of their direct bearing on the economic factors involved.

". . . the first step is removal of political obstacles and legislative threats."

This committee includes in its membership C. C. Carlton, secretary, Motor Wheel Corp., who has been a member of the S.A.E. since 1917.

ELEVEN automobile manufacturers who had a combined sales volume of $309,614,000 in 1933 netted a combined profit of 1.2 per cent on their operations, according to a compilation issued recently by the National City Bank of New York.

The average per cent of profit as related to dollar volume of sales of 240 companies in many industries analyzed was 0.9 per cent. Cotton mills with a 7.1 per cent profit and manufacturers of drugs and sundries with 7.0 per cent profit lead the list. At the bottom came iron and steel makers with a deficit of 6.9 per cent.

"These figures are of particular interest," the accompanying review states, "in view of the charge, frequently repeated, that capital exacts undue tolls upon the products of industry."

TWENTY *Years Ago This Month:* Entertainment features projected for Cape May Summer meeting scheduled for June included entertainment of visiting members by "players" from the Sections; a regular banquet tendered with the compliments of the hotel management; an illustrated lecture describing main features of the coming Second European Visit of the Society consisting of stereopticon slides and moving pictures of points of interest and a talk by Orrel A. Parker; and "for the kiddies, 'movies,' showing, of course, such pictures as will appeal particularly to their juvenile emotions."

"The truck standards division," wrote J. E. Hale, experimental engineer of Goodyear, "in its efforts to arrive at a universal schedule of solid tire carrying capacities satisfactory to all concerned, discovered that the situation is really quite complex. . . ."

Analysis of geographical distribution of S.A.E. members showed Michigan with 261 full members, New York with 226, Ohio with 155, and Pennsylvania with 101. Total number of full members was 1154; of associates, 530; and of all grades combined 1807.

New Group Tackles Tractor and

QUALIFYING in every respect to the homely phrase, "Getting Down to Brass Tacks," which the official program advanced as the convention slogan, the first meeting sponsored by the recently revived National Tractor and Industrial Power Equipment Committee of the Society of Automotive Engineers, held in Milwaukee April 18-19, not only attracted an attendance far exceeding expectations but aroused a degree of interest and developed so many new angles to the problems of this group that a rich field for discussion was opened which undoubtedly will lead to more and frequent meetings.

As chairman of the first session, Paul W. Eells, LeRoi Co., Milwaukee, and chairman of the Milwaukee Section of the Society, spoke prophetically: "On the roster of the Society you will find the names of many engineers affiliated with companies making a product which apparently has no connection with the automotive industry. This is particularly true of the territory covered by the Chicago and Milwaukee Sections. However, we believe that if the eastern Sections were to put on one Section meeting each year devoted to industrial power equipment, they would find a considerable amount of interest. The reason, as we see it, for this interest in the Society among engineers in remotely related industries is that the Society comes nearer than any other engineering organization to filling a certain large void that has existed between engineering societies. We believe the time has come when this group of engineers would like to get together to talk over their troubles and achievements; we believe they are entitled to an activity of their own and we hope this will be the beginning of this activity. It is even possible that we as a group might contribute something to the improvement of the automobile."

Fowler McCormick is First Speaker

The meeting began with a paper on "The Relation of Engineering to Manufacturing and Distribution in the Farm Implement Industry" by Fowler McCormick, International Harvester Co., who, it may be said incidentally, appraised the importance of the meeting so highly that he remained from start to finish of all of the four sessions. Admitting frankly that one of the main aims of his paper was to stimulate discussion, Mr. McCormick emphasized "the idea that the relation of engineering to manufacture is that the engineer shall design machines which can be well and economically manufactured, and that the relation of engineering to distribution is that the engineer shall design machines which are readily saleable. In order that the engineering department may fulfill its true function, and that the most satisfactory results may ensue, it would seem as if the three following general conditions must prevail:

"1. Engineering department personnel must be in close touch with the field and know trade conditions and the farmer's requirements. It must also understand manufacturing well enough to know how to design machines for manufacturing.

"2. It is of great importance that as close a personal contact as possible exist between the members of the engineering and sales departments, and the engineering and manufacturing departments.

"3. For proper functioning there must be an understanding on the part of each department of the point of view and problems of the other departments, and a sincere desire to cooperate with and to assist each other."

H. B. Dinneen, vice-president, Minneapolis Moline Power Implement Co., was not able to be present but submitted a written discussion which said in part: "Mr. McCormick in his paper bears on the very fundamentals that create success in an industry like ours. His considerations not only affect the relations of three very important departments but have to do with management itself. And this is right and proper, for where independent departments ignore the principles and policies that go to make a successful enterprise, organization breaks down and failure is the result."

H. H. Howard Opens Second Session

Informal discussion of Mr. McCormick's paper also was in close agreement. It developed many angles far afield from the supbject, boring its way even into the merits of rubber-tired tractors for the farm. Discussion of rubber-tired tractors focused attention on the fact that while the automobile industry provides a plush seat for the farmer in his passenger car for use a few hours each week, the tractor industry apparently had done nothing to provide a comfortable tractor seat which the farmer occupies ten hours a day, six days a week. As to rubber tires, all of the speakers were in agreement that such a tractor has its limitations, but seems to be doing a good job and is desirable and competent for 85 per cent of the farm jobs it is required to do.

With J. B. Fisher, chief engineer, Waukesha Motor Co., in the chair, the second session was opened by H. H. Howard, Caterpillar Tractor Co., with a paper on "Some Diesel Tractor Problems" which likewise provoked a splendid discussion. Mr. Howard said that the Diesel engine's chief claim to consideration for tractor service lies in its recognized ability to produce power economically. This economy is obtained first, because through its high compression the Diesel consumes less fuel per unit of power produced than the so-called semi-Diesel engines or other types of conventional oil engine; second, because the Diesel operates most satisfactorily on a wide range of cheap petroleum fuels.

Saying that no Diesel tractor business in reasonable volume may be expected unless the engine will render service as generally satisfactory as the gasoline engine, Mr. Howard pointed out the efforts of his company to meet the problem set up by the fact that, as distinguished from fuel economy demand in Europe and other countries, the American farmer will not tolerate any unusual maintenance or operating difficulties. To this end fuel injection pumps and fuel injection valves have been made entirely interchangeable as individual units, and all field adjustment, calibration or timing of these parts is entirely eliminated. Worn-out equipment is exchanged for new on a flat-rate service exchange basis.

Mr. Howard also emphasized that if the Diesel engine is to continue to deliver the full economic advantages that its cycle makes possible, it must not be temperamental as regards fuel.

Since eight or nine tractor manufacturers and five or six major oil companies were respresented at the meeting, Mr. Howard's paper was quite thoroughly dissected. C. B. Jahnke, research engineer, International Harvester Co.,

Unit Power Equipment Topics

pointed out that the Diesel engine costs about three times as much as the gasoline engine of comparable size, while maintenance on Diesels runs approximately 50 per cent more. Mr. Howard replied that maintenance need not necessarily be higher, but admitted that in 50 per cent of the Diesel tractors in the field maintenance does run higher. This is due, he said, entirely to the personal element in owner-operator. Representatives of oil companies contributed to the discussion by showing that there is no immediate danger of fuel shortage or possibilities of premiums, pointing out that the matter simmers down to a matter of economics.

The other side of the tractor power picture was presented by E. R. Jacoby, engineering department, Continental Motors Corp., whose subject was "Spark Ignition Engines for Agricultural and Industrial Use". The paper covered study and research in response to requests for operating data of a widely diversified character, for example, on engines using natural gas, butane, gasoline of 80, 70 and 60 octane rating, kerosene and fuel oil.

With the aid of lantern slides, Mr. Jacoby explained results, adding: "Inspection of the curves shows that from the standpoint of first cost, interest charges and fuel operating costs at prevailing fuel prices, the butane carburetted engine is an outstanding buy for any number of hours of operation per year. The next cheapest job is the Diesel when operated in excess of 1200 hours per year, although its upkeep expense proves much greater than that of a carburetted fuel oil engine and the saving might easily be nullified. The next cheapest job to own is the carburetted fuel oil engine. This looks like a 'natural' for the farmer and industrial power user who operates less than full time.

In rebuttal to the arguments of the Diesel school, Mr. Jacoby expressed the opinion that the stresses set up in an injection engine will always be far beyond those in carburetted spark ignition engines, judging by the vast amount of experimenting his company has done in both fields.

Because of the inability of H. C. Dickinson, chief, power plant section, U. S. Bureau of Standards, and past president of the Society, to leave Washington, H. C. Merritt, general manager, tractor division, Allis-Chalmers Mfg. Co., presided over the evening session. This was devoted entirely to a paper on "Research in Agricultural Engineering" presented by Dr. S. H. McCrory, chief, bureau of agricultural engineering, U. S. Department of Agriculture. He predicted that despite thoughts to the contrary, mechanical power will play a much larger part in American agriculture than at present. He explained that agricultural engineering is not a matter of designing equipment but rather of finding out what equipment is necessary and desirable to meet agricultural problems.

A. W. Lavers, chief engineer, Minneapolis Moline Power Implement Co., was chairman of the next morning's session, with A. C. Staley, research engineer, Chrysler Corp., as principal speaker. The subject was "Requirements of Tractor and Industrial Engines". After discussing the prime requisites of both the spark ignition and the Diesel engine, Mr. Staley said: "In any event, each adaptation should be studied and the load factor, hours of operation, fuel cost and other items should be laid out so as to determine the comparative operating costs of the two types of prime movers. The injection engine will make its best showing when the load factor is high, the hours are long and the differential cost between gasoline and a suitable fuel for its operation is high." He added that the development of oil-burning engines of both types has been handicapped somewhat due to insufficient knowledge as to fuels available and their economic and technical characteristics.

Fuels also formed the principal topic at the final session, at which C. G. Krieger, agricultural engineer, Ethyl Gasoline Corp., and general chairman of the convention, presided. "Chemical Hay for Mechanical Horses" was the title of a most interesting paper presented by Dr. R. E. Wilson, vice-president in charge of research and development, and D. P. Barnard, assistant director of research, Standard Oil Co. (Ind.). This dealt largely with the present and future price situation of tractor fuels with the intention of setting forth facts that might assist tractor engineers in designing engines with relation not only to cost of fuels but their future availability. Under present economic conditions, No. 1 furnace (fuel) oil was pointed out as the most economical fuel available for tractors, and the most desirable as well. A cracked product is bound to be cheaper than a straight run product. Moreover, No. 1 fuel oil has not been reached by federal, state or local taxes, such as are imposed on all grades of gasoline, although the tax factor is somewhat minimized by the system of refunds of gasoline taxes made by some states. The resulting discussion developed the fact that less than 10 per cent of the tractors are used in States not making refunds on farm use.

All sessions were held at the Pfister hotel in downtown Milwaukee.

What **Tractor Talkers** Said

FOWLER MCCORMICK:

"The personnel of the engineering department must be in close touch with the field and know trade conditions. . . ."

H. H. HOWARD:

"If the Diesel engine is to continue to deliver the full economic advantages that its cycle makes possible, it must not be temperamental as regards fuel."

E. R. JACOBY:

". . . from the standpoint of first cost, interest charges and fuel operating costs at prevailing fuel prices the butane carburetted engine is an outstanding buy for any number of hours of operation per year. . . ."

A. C. STALEY:

"The injection engine will make its best showing when the load factor is high, the hours are long, and the differential cost between gasoline and a suitable fuel for its operation is high."

The Near East Enters the Petroleum

Boats passing the Maud Bridge (pontoon bridge) in Baghdad. This is the only type of "permanent" bridge at Baghdad. Sections of the pontoon bridges are swung upstream to permit boats to pass. The sailboat is the typical native craft used throughout the Levant

OPENING of a new oil field is always news for the petroleum industry. When the new producing field also ushers a new nation into the world, the event takes on historical proportions.

During the present year Iraq became an independent nation and also one of the greatest economic influences in the petroleum industry. The Book of Daniel records the trial of the three Hebrew youths in the fiery furnace. If tradition can be trusted this was none other than the "eternal fires" which burn today near the Baba Gurgur well.

North of Baghdad made famous by Harun ar Rashid, and East of the Tigris River, is the town of Kirkuk. Here are centered the oil producing activities of the Iraq Petroleum Co. It is directly on one of the oldest trade routes in the world, and in the path traversesd by Genghis Khan, Alexander the Great, and the 10,000 Greeks, as these conquering hordes passed back and forth between Europe and Asia.

Two conditions have contributed to the fact that, in spite of this area being the first in the world where petroleum is known to have existed, it is one of the latest to attain development. The first of these factors is geological. In spite of early evidences of existence of petroleum the structural geology of this area is difficult to read. Oil is found there in the limestone of the Miocene age. The formations range from Triassic to Recent. Surface indications are misleading. At Kirkuk there is only a monocline, with northeast dips visible on the surface, but drilling has developed a perfect anticline underground.

The second factor in the problem of developing this field was transportation. To take bulk crude down the Tigris to Abadan and then reship by tanker through the Persian Gulf and around the Arabian Peninsula through the Red Sea, the Suez Canal and the Mediterranean to London or Hamburg represents a journey of nearly 6600 miles. It is only 5000 miles from Galveston, Texas, to these same ports. If a pipe line could be constructed across the Syrian desert a distance of about 600 miles, then, using the (Syrian) port of Tripoli as a shipping point, the distance to London or Hamburg is only 2400 miles.

The Iraq Petroleum Co. was formed to develop this field and to build the pipe line so that the oil could be marketed economically. This company represents an international group of oil interests which are cooperating in the development. These are the Anglo Saxon Petroleum Co. (Royal Dutch Shell Group), D'Arcy Exploration Co. (Anglo-Persian Group), Compagnie Francaise des Petroles (French Group), Near East Development Corp. (American Group), and Participations and Investments Ltd. (Gulbenkian Interest).

The Iraq Petroleum Co. has received a grant from the Iraq Government which permits it to take, manufacture, transport and sell petroleum products for a period of 75 years. This is limited to a definite geographical area including the vilayets of Baghdad and Mosul from the East bank of the Tigris, North to the Iraq-Turkish border and east to the Iraq-Persian border. The company also agreed to construct the pipe line and to accept and pay for a minimum quantity of crude oil each year upon a royalty basis. At the end of the 75-year period the agreement is terminated and all of the Company's lands, buildings, pipelines and other facilities revert to the Iraq Government without charge. The Chairman of the Company is Sir John Cadman who is one of the leading British executives in the petroleum industry. Under his capable leadership the construction of this great project is now nearing completion.

The actual project was placed in charge of H. S. Austin, who, following his association with the Tuscarora pipeline, built the Ajax pipeline, a 400-mile double 10-in. line from Glen Pool, Okla., to Wood River, Mo. Associated with

[This paper was presented at a meeting of the Indiana Section, Oct. 12, 1933.]

Industry — —

By A. W. S. Herrington
President, Marmon-Herrington Co., Inc.

Mr. Austin and in charge of all pipeline work in the field is Capt. M. M. Stuckey, formerly general manager of the Andean National Corp. Ltd., Colombia, and who will be in charge of operation of the Iraq pipeline and loading projects on completion of construction.

Associated with these two has been James J. Jameson, C.B.E., one of the Anglo Persian Oil Company's outstanding men, formerly their pipeline superintendent in Persia and afterwards resident director and general manager. He has brought to the project unrivalled first-hand experience in that terrain.

The pipeline starts at Kirkuk and runs as a double line in a southwesterly direction. It crosses the Tigris River at Fatha Gorge and the Euphrates River at Haditha. A few miles beyond this point the northern branch tends to a westerly direction, crossing the Iraq-Syrian border south of Abu Kemal then through Palmyra, Furklus and Homs to the seacoast at Tripoli, Syria. The Southern branch goes in a southwesterly direction down the Wadi Hauran to Rutbah and at the Jebel Asfar turns due west through Mafrak, then down across the Jordan Valley into Palestine and to the sea at Haifa. The northern line from Kirkuk to Haifa is 618 miles long.

The main pipeline is 12-in. interior-diameter steel pipe. All joints are being electrically welded. The 1149 miles of pipe amount to approximately 120,000 tons of steel. The line is being buried in the ground throughout its entire length and for additional protective purposes it is being coated and wrapped.

The telephone lines alone represented a construction project of no mean proportions. About 1140 miles of overhead line, 25,000 tubular steel poles with cast-iron socket bases, 25,000 steel cross-arms, 120,000 spindles and insulators, and over 4000 miles of cadmium-copper telephone wire.

Due to the peculiar effect of the water of the Tigris and Euphrates upon submarine cables it was necessary to construct steel towers 130 ft. high to carry the wires across these rivers.

There are pumping or booster stations located at about 60-mile intervals. Diesel engines are being used to drive the booster pumps. It will require about 45 500-hp. engines for this purpose. These engines are capable of operating on crude oil drawn from the line.

In addition to all the construction dumps and stations with the multiplicity of buildings and equipment involved, all of the permanent buildings and equipment for the pumping stations must be moved into the desert and assembled.

The material for the construction of the pipeline was transported to three seaports: Haifa, Palestine, Tripoli, Syria, both on the Eastern end of the Mediterranean, and through the Suez Canal and Red Sea to Basra on the Persian Gulf.

All material landed at Tripoli, Syria, was conveyed inland by the Syrian Railroad to a concentration dump at Homs, Syria. From here it was transported into the desert by trucks. The economical haul point was found to be at the Wadi Rutka near Abu Kemal and on the Syrian-Iraq border.

On the southern line the material was landed at Haifa and transported inland to Mafrak by the Palestine Railway. From here it was transported into the desert by motor truck to the economical haul point at H-3 near Rutbah. This section included the infamous lava country, an old volcanic section, where for over 100 miles the ground is strewn with a mass

Native Kurdish village near Borsivi near the northern Iraq border. These mountain villages are found all along the mountain slopes near springs or streams.

Typical Arab ferry. This one is at Rawa on the Euphrates just north of Ana. The men in uniform are Iraqi police. Across the river in the right background are two of the famous "water wheels" used for irrigation along the Euphrates between Aleppo and Hit

of boulders varying from the size of a man's head up to five and six feet in diameter. Transportation through this area was exceedingly difficult.

The material from the Persian Gulf was brought up from Basra through Baghdad to Kirkuk over the Iraq State Railway. From Kirkuk material was transferred by truck down to the banks of the Tigris River at Fatha Gorge. Other material was brought up the West bank of the Tigris by the Iraq State Railway and delivered to Baiji Station. This was distributed by motor truck between Baiji and the East bank of the Euphrates at Haditha.

A concentration dump was established at Haditha to collect all material for construction of the line westward from Haditha out to Wadi Rutka on the north line and down the Wadi Hauran to Rutbah on the south line.

A ferry to transfer this great tonnage across the Euphrates at Haditha was not practical on account of the uncertain water conditions on the river. Some idea of the difficult water conditions can be gained from the fact that on the Tigris River there is a 23-foot difference between high and low water. Some other means insuring reliable transportation at all times had to be adopted.

The solution was the use of a blondin or overhead cableway. The Euphrates span is 1830 ft. and the cable towers were 140 ft. high. The load capacity at top speed is five tons with a lifting speed of 240 ft. per min. and a transverse travel speed of 960 ft. per min. At slower speeds 10 tons can be handled.

The main ropes of this cableway were 6¾ in. in circumference. The travel ropes were three inches and the hoist ropes were 1¾ in. Loads can be carried on flat platforms or in wire slings.

From Haditha into the desert it was again necessary to use trucks. The movement of these thousands of tons of material across a 600-mile stretch of trackless desert is one of the most outstanding feats of motor-transport work ever undertaken. It can be conservatively stated that this line would not be a feasible project except for the efficiency of the motor trucks available today.

It must have shocked the Arab caravan leaders to see great truck-tractors hauling 50-ton loads of steel pipe at 35 miles an hour, out across terrain which has previously only known the pad-like feet of the camel caravan.

It was not until 1923 that it was known to be possible to cross this arid area by motor-vehicle. An enterprising New Zealand Army officer who had served with General Allenby during the Mesopotamian Campaign was the first to prove that the feat was possible. His name is Norman Nairn, and today the Nairn Transport Co., operating a bus and freight line from Damascus to Baghdad forms a motor line over one of the Old World tradeways.

Year in and year out, commencing in the post-Armistice period and in the face of armed insurrection and adverse climatic and ground conditions, Nairn's convoys move back and forth between the Mediterranean coast and Baghdad carrying mail, passengers and express. He has been a real pioneer and was the first to introduce and operate a Diesel-engined truck for transport work in this desert country.

The discovery well of the Kirkuk Field was struck on Oct. 14, 1927, at Baba Gurgur near Kirkuk. Since that date the I.P.C. has concentrated on that area and delimited a field promising enough to justify the 84,000 bbl. pipeline. The wells are relatively shallow, averaging about 2700 ft. deep and are fairly easy to drill. The producing formations are limestone so that after the producing string of casing has been set there are no caving difficulties requiring liners or screens.

Although no very continuous flow tests have ever been made, excepting on the discovery well which flowed for 8½ days through 10 in. casing, it is believed that sufficient wells have been drilled to initially supply the pipeline. The crude obtained is light with a Baumé A.P.I. gravity of 35 to 36 deg.

The oil from the field is concentrated at Kirkuk and here it enters the pipeline for transportation to either Tripoli or Haifa. By Dec. 31, 1935, the line will be complete. It has been designed to be capable of handling about 30,000,000 bbl. of oil a year.

The marine transfer and loading stations at Tripoli and Haifa were also exceedingly difficult engineering projects. The seacoast is shelving and shallow and boats must stand off shore at least a mile and load from buoyed flexible lines. The situation at Haifa is being simplified by a harbor-construction project which is nearing completion.

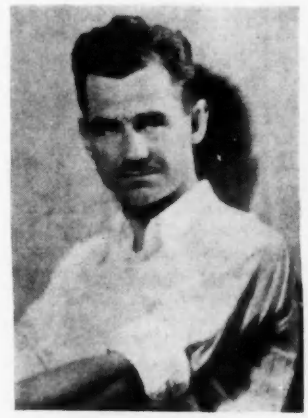
Iraq seems to have definitely taken its place in the list of oil-producing areas of the world. As a newcomer to the world family of nations, oil is its principal natural resource and source of national income. Its strategic location with reference to the markets of Europe, and particularly to those countries bordering on the Mediterranean, will guarantee a ready market for the crude.

It is tragic that H. M. Feisal ibn Hussain, King of Iraq, could not have lived to see his country realize in full his dream of self-supporting autonomy. His thoughtful administration developed a well-organized system of government, to which is now added the independence of assured national income. Surely this is all a fitting memorial to a great life well lived.

He passed from a world torn by industrial strife, with sectional and national interests seeking to control commercial policy for individual selfish ends. This great project representing the joint effort of three nations and several great petroleum companies is an excellent example of constructive cooperation which might well be studied by all industrial executives.

At the Summer Meeting
Saranac Inn, June 17-22

Things Will Happen!



Lowell Thomas

LOWELL THOMAS

will be there in person. He will talk at the session on Thursday evening, June 21. He has many an arresting tale to tell—tales of adventure in far-off lands where he has not only been a witness to great events, but has time and again surprised history in the making and played a part in the momentous drama.

TECHNICAL SESSIONS

will be held every morning and every evening. A new high point in quality of papers is predicted by Meetings Committee chairman Alex Taub. Final selections of papers are now being made.

FULL PROGRAM

There will be more technical sessions than at any previous summer meeting.

WATER CARNIVAL

will be the entertainment feature on Monday afternoon. Fun for spectators and participants is guaranteed because the event will be under the direction of S. S. Dickey who at the last Saranac gathering operated such successful events. He will also be the master of ceremonies for the

FIELD DAY

which will keep the members and guests in good spirits on Tuesday afternoon.

TRANSPORTATION REVUE

is an entirely new feature and consists of a special review of new cars and other automotive vehicles, providing an opportunity for riding and driving as well as for careful inspection under favorable conditions.

TENTATIVE PROGRAM OUTLINE

Sunday evening, June 17: Business session—general session—something to interest everybody.

Monday: Morning sessions—passenger car (That Sound Problem), transportation and maintenance (Accident Prevention). Afternoon—water carnival. Evening—truck, bus and railcar (Light Trucks).

Tuesday: Morning sessions—transportation and maintenance (Maintenance Costs) truck, bus and railcar (Railcars). Afternoon—field day. Evening—passenger car (Streamlining).

Wednesday: Morning sessions—Diesel (Design and Development) passenger-car body (Basic Principles of Design). Afternoon—fuels and lubricants (Light Crankcase Oils). Evening—fuels and lubricants (Aircraft and Diesel Fuels).

Thursday: Morning sessions—passenger car (Engine Symposium). Aircraft (Controllable-Pitch Propellers). Afternoon—aircraft engine (Combustion and Design Research). Evening—general session (Lowell Thomas and the Grand Ball).

Friday: Morning sessions—aircraft (Towing Basin Contribution to Research) aircraft engine (Operation Under Extreme Conditions).

What Members Are Doing

Dr. George W. Lewis, Clarence D. Chamberlin, Edgar S. Gorrell, and *Maj. Gen. Benjamin D. Foulois,* are members of the Society serving on a commission of 11 headed by Newton D. Baker to study the operations and efficiency of the Army Air Corps. The commission was named by George H. Dern, secretary of war, and includes 6 civilians and 5 Army officers.

Roy E. Cole has joined the engineering staff of the Hupp Motor Car Co. Mr. Cole was chief engineer of the former Durant Motors, Inc., and later of the Rockne Division of Studebaker. Recently he has been operating the Cole Engineering Co., Detroit. He has been a member of the Society since 1928.

Dean M. Gillespie is president of the Motoroyal Oil Co., Denver, Colo. He was district manager in Denver for the White Co. prior to his new connection.

John A. Leahey is general manager of the Leahey Motor Car Co., Irvington, N. J. Formerly he owned the Leahey Marmon Co. and the Cape May Chemical Co., Tuckahoe, N. J.

S. Edward Rowe is service engineer in Cincinnati for United Motors Service. He was chief engineer of the Stutz Motor Car Co. of America, Indianapolis.

Elwin S. Titus is engineer in charge of installation of mechanical and electrical equipment at the Pioneer Mine, McCormick, S. C. The mine is operated by the Carolina Exploration Co., Inc.

L. C. Welch

L. C. Welch, who has been manager of the lubricating department, Standard Oil Co. (Indiana), has been appointed assistant general manager in charge of the lubricating and technical departments.

George J. Liddell is test engineer in charge of automotive tests for the Doherty Research Co., Long Island City, N. Y. He was research assistant at Purdue University, Lafayette, Ind.

Edwin L. Smith is full-size-body layout draftsman with the Briggs Mfg. Co., Detroit. He formerly held a similar position with the Pierce-Arrow Motor Car Co., Buffalo.

Merritt A. Mieras, formerly with the Holley Carbureter Co. as a sales engineer, is now engineer in the experimental department, Bohn Aluminum & Brass Corp., Detroit.

(Continued on page 32)

Franklin F. Chandler

FRANKLIN FAY CHANDLER, vice-president in charge of sales, Ross Gear & Tool Co., a very active and prominent member of the Society, died April 6 at Lafayette, Ind. His death followed an illness of several weeks. He was 57.

Mr. Chandler was widely known as an automotive engineer and enjoyed the friendship of many executives in all branches of the industry. He joined the Society in 1919 and began wide participation in its affairs when he became Chairman of the Indiana Section in 1923. In 1926 and 1927 he was a member of the Council and has held membership or chairmanship of various committees of the Society on 29 occasions since 1923.

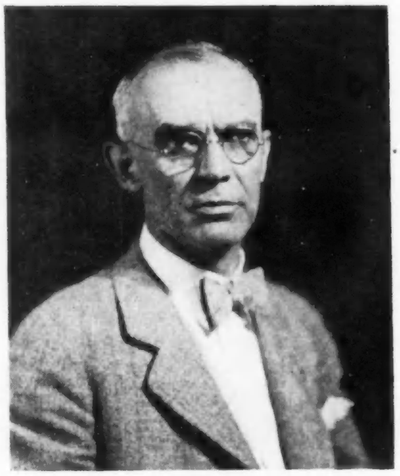
The breadth of his engineering interests may be judged from the fact that he was a member of the Sections Committee (Chairman), the Research Committee, the Military Motor Transport Advisory Committee, and the Truck, Bus and Railcar Activity Committee, in addition to numerous subcommittees.

Mr. Chandler was born in 1876 at Indianapolis. He took a mechanical engineering degree at Purdue University, Lafayette, Ind., and for 18 years was connected with the Chandler and Taylor Co., of which his father was one of the founders. He served his Alma Mater as alumni trustee and in June 1916 became chairman of its Standing Committee in Engineering and Technology.

From 1922 to 1924 he was manager of sales for the Chandler & Taylor Co. In this period he helped to organize and became president of the Adaptable Tractor Co., Indianapolis. In 1924 he became chief engineer, Ross Gear & Tool Co., Lafayette, Ind., and in 1928 was elected a vice-president of the company. In 1933 he took over its sales activities as vice-president and sales manager.

Mr. Chandler is survived by his wife, and four children.

The Society was represented officially at Mr. Chandler's funeral by A.W.S. Herrington, president, Marmon-Herrington Co., Indianapolis. The funeral was held at Lafayette, Ind., and was reported as a "simple memorial service, in its nature characteristic of Mr. Chandler's life."



F. F. Chandler

A Tribute to Fay Chandler

THE automotive industry and the S.A.E. have suffered a great loss in the death of Fay Chandler. To those who knew him well, and they were many, for he had a gift for friendships, he was a good friend, a delightful companion, a thoroughly capable engineer, and a manufacturer respected for the principles underlying his business policies as well as the quality of his products, and a gentleman in all that the true meaning of the word implies.

A visit from "Fay" was always a bright spot and a relief from the wearisome round of a business day.

I know that I express the sentiments of his many friends in the industry and in this Society when I say that his absence from our gatherings will be deeply felt. To his bereaved family we reverently bring our tribute of sympathy and sorrow.

**D. G. Roos, President,
Society of Automotive Engineers**

The Cowling and Cooling of Radial Air-Cooled Aircraft Engines

By Rex B. Beisel, A. Lewis MacClain and F. M. Thomas

THIS paper presents the results of coordinated research by The Pratt & Whitney Aircraft Co. (engines), the Chance Vought Corp. (airplanes), and the U.A.T. Research Division, all subsidiaries of the United Aircraft & Transport Corp. These studies were directed toward improving the performance of airplanes through reducing the drag of radial air-cooled powerplant installations as nearly as possible to the minimum necessary for adequate cooling.

The studies were supported by a considerable amount of experimental data. Extensive wind-tunnel tests provided quantitative measurements of airflow and drag for many combinations of baffles and cowling, and throughout the whole work simultaneous flight-tests checked results and contributed to the final conclusions. The successive stages of baffle development, as well as the experiments with various sizes and shapes of cowling, are discussed. The optimum combination ultimately found is described in detail.

The results of the studies show that much can be done to improve the arrangements now in general use. The use of the best type of baffle developed during the study, together with cowling designed in accordance with limitations determined during the course of the research, showed marked improvement in both aerodynamic performance and cooling over those of conventional arrangements, as is evident from the data and discussion presented.

The paper also points out the desirability of controlled cooling to compensate for variations in temperature and operating conditions, comparable to the use of a retractable radiator. The development of an entirely new type of adjustable cowl which successfully accomplishes this important objective is explained. Finally, mention is made of a few of the more important changes in powerplant installation which are made desirable by the use of the new baffling and cowling system.

THE trend of design in the modern airplane has been toward improved performance realized through external cleanness. It is apparent that the number of essential units comprising a modern airplane is nearly a minimum at the present stage of the art, and it appears also that the possibilities of further striking reductions in the drag of these units, due to change in form or shape either individually or in combination, are not great.

Other than through detail refinement, what can be done to further reduce parasite drag? Where, in the airplane, are we sacrificing performance through ineffective design or operation? What other drag items of the airplane can be studied and investigated with possibilities of improved efficiency? A careful study leads us to the powerplant and the drag introduced in properly cooling the engine. We know that the total power expended in cooling is greatly in excess of that actually required to take care of the skin friction resulting from passing air over the cooling surfaces of the radiator or of the cylinder fins. It has been shown[1] that in an actual case the total power expended to cool an engine equipped with a Townend ring was some ten times greater than that which would be used in the ideal case, where the only air resistance realized would be that of the air "wiping the heat" off the cooling-fin surface. Since, as pointed out, the engine spent "16 per cent of its power in keeping itself cool," it is quite clear that we have some real possibilities to work with in endeavoring to improve the efficiency of the cooling system and thereby realize a marked reduction in the parasite drag of the airplane.

[This paper was presented at the 1934 Annual Meeting of the Society, Detroit. Mr. Beisel is connected with the Chance Vought Corp.; Mr. MacClain, with The Pratt & Whitney Aircraft Co.; and Mr. Thomas, with the Research Division of the United Aircraft & Transport Corp.]

[1] See *Aircraft Engineering*, February, March and April, 1932: The Theory and Practice of Air Cooling, by D. R. Pye.

Fig. 1—Pratt & Whitney Aircraft Co. "Twin Wasp" Engine without Ring Cowl and Baffles

The purpose of this paper is to discuss the results of the experimental work which has been carried on jointly on the problem of drag reduction and controlled cooling of radial air-cooled-engine installations in aircraft. The tests and experiments have been conducted over a period of more than two years and are still continuing. They cover the development of both engine baffles and engine cowling. The main objective was to obtain the greatest possible improvement in engine cooling and airplane performance by combining effectively the design of both baffles and cowling. In attacking the problem of improving the cooling of the radial-engine installation, it may be in order to point out in a general way two definite controlling factors which must be considered.

First, we must accept a definite air resistance to obtain engine cooling, irrespective of the arrangement used in realizing this cooling. This, as previously stated, is the resistance which must result in "wiping off" a certain amount of heat from the cooling fins. It is predicated on the cooling phenomenon of removing heat from metal surfaces. In actual practice the resistance resulting from the removal of this heat is considerably greater than that due alone to the skin friction over the fins. This increase results from change of path, interference, restrictions, and turbulence of the cooling airflow from the time it enters the front opening, passes the engine and is again directed to the outside airstream. It is quite apparent, however, that the nearer we approach this necessary minimum drag, through the method employed in directing the cooling air to pass over the fins and in leading this air away from the fins, the nearer we come to obtaining necessary cooling with the minimum loss in airplane performance.

Second, we must accept a certain air resistance due to the parasite drag resulting from forcing the engine through the air. This assumes, of course, the arrangement of the engine in the conventional airplane. The engine has a certain frontal area; the propeller is in a given location. The best we can do, even assuming no airflow for cooling, is to house the engine in the best flat-nosed streamline shape and accept the resulting parasite drag.

With these two basic facts in mind, it is apparent that the real problem ahead of us is so to combine the design of engine cowling and the method employed in directing and controlling the airflow over the fins as to result in adequate engine cooling and, at the same time, minimum airplane drag. In other words, our objective is to obtain a combined arrangement such that the power expended in circulating the cooling air approaches as nearly as possible the ideal minimum, and that the power expended in forcing the engine through the air approaches that of an ideal streamline shape enclosing the engine. Following are some of the items of design which must be considered:

An opening must be provided in the nose of the outer engine cowling for entrance of cooling air. How large should the opening be? What is the best shape of the nose of the outer engine cowl? Should a crankcase cowl be provided? The cooling air must be directed to the cylinder cooling fins. If baffles are used, what size and shape will be most effective to give maximum cooling with minimum drag? The air leaving the cooling fins must be carried away and directed back into the airstream. What shape of engine accessory or wrapper cowl will give the best flow-path? What is the best shape of the after portion of the outer engine cowl? How large an exit opening should be provided for the cooling air? How can this cooling air be combined with the outer airstream to result in minimum turbulence? The cooling requirements in a given installation vary with operating and with seasonal conditions. Can a means be devised to control the cooling to meet these variable conditions?

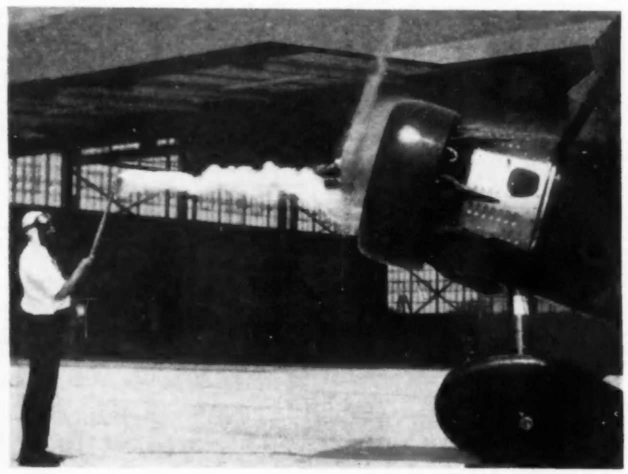


Fig. 2—Checking with a Smoke Candle the Airflow to a Pratt & Whitney Aircraft Co. "Twin Wasp" Engine Equipped with a Ring Cowl But without Baffles

It readily can be appreciated that the cooling-drag problem, with its many possible combinations of variables, was difficult to attack in a logical sequence of tests. The experiments leading to the final conclusions covered a great many flight tests and an even greater number of wind-tunnel tests, in the process of which we learned something first about one combination and later about another. It was necessary to study each test as we varied the set-ups, in order, logically, or, in some cases as we found later, illogically to proceed with further testing. For this reason, the following discussion, rather than covering the time sequence in obtaining results with various combinations, covers the work done and final results obtained in the development of:

(1) A system of cylinder baffling

(2) Engine cowling suited to the baffle system

(3) Controlled cooling by means of the adjustable cowl flap

(4) Other items of powerplant installation incident to items (1), (2) and (3)

Cylinder Baffles:—Important work leading toward our new conception of baffling was done in June, 1931, in connection with flight tests of the original Pratt & Whitney Twin Wasp engine (two-row, 1830-cu. in. displacement). The standard baffle in use at that time was the conventional V-type skirt-baffle developed for use on single-row engines. Obviously, the same baffles were not adaptable to the front row of cylinders. Further, in developing a new baffle system, the problem was made doubly difficult by the fact that the cooling of the front and rear banks was dependent one on the other. A solution had to be attained wherein the cooling of either row did not adversely affect the cooling of the other row.

Excerpts from these early reports show clearly the difficulties encountered and the gradual approach to the final solution, as follows:

"These cowling and cooling tests on the Vought V-50 with two-row radial engine No. X-26 installed were made prior to the airplane performance-tests in an effort to secure satisfactory engine cooling together with as high a top speed as possible.

"With no ring cowl, the cooling of the engine was very satisfactory, both in level flight and in climb. In nearly all cases, the latter was the critical condition. (See Fig. 1.)

"The addition of the narrow ring cowl to this set-up increased the top speed by 12.5 m.p.h., but raised the engine temperatures to prohibitive values. Those of the front heads were increased by an average of 115 deg., front bases by 55 deg., rear heads by 73 deg., and rear bases by 24 deg. fahr.

"These tests illustrate the object of all the succeeding experiments; namely, to provide a narrow ring or N.A.C.A.-type outer cowling, which would give this very appreciable increase in maximum speed and at the same time to devise a means of getting permissible engine operating-temperatures.

"With the nose and front inter-cylinder cowling removed but with the ring cowl still in place, the change in temperatures was negligible. Speed-course runs made on Aug. 17, and Sept. 8, 1931, showed that the variation in top speed due to this change was less than the experimental error.

"For tests Nos. 14 and 15 the standard V-baffles on the rear bank were removed, leaving the narrow ring in place. As a result, the rear cylinders, both head and base, ran very much hotter, while the front cylinders were only slightly cooler. Throughout these flight tests the V-baffles were found to be the only satisfactory means of cooling the rear bank. With them in place, the front bases were critical and it was with these front bases that most of the remaining experi-

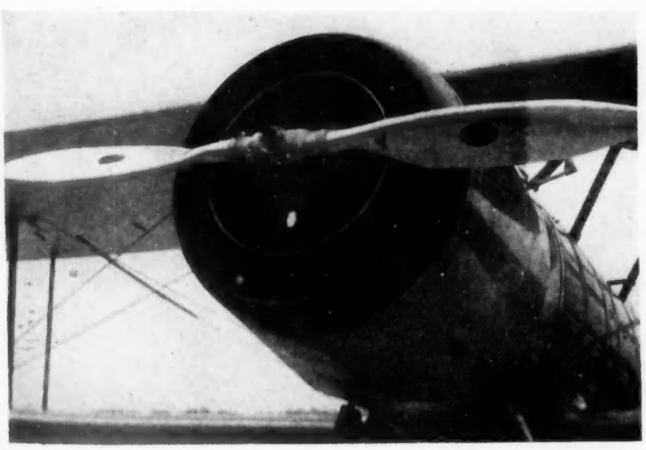


Fig. 3—A Full-Length N.A.C.A. Cowl and Inner Flange Cowl

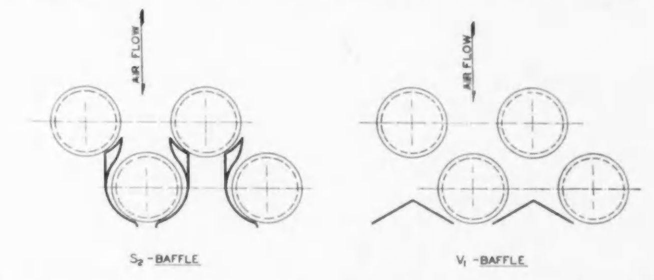


Fig. 4—Comparison of S_2 with V_1 Baffles

ments were concerned. At this point some tests of the airflow through and around the narrow ring cowl were made. With the engine turning up on the ground, a smoke canister was held about 6 ft. in front of the propeller, (as shown in Fig. 2). The smoke showed a decided tendency to build up in front of the ring cowl and then flow out around it. Apparently very little was passing through the engine. Back of the trailing edge of the ring cowl a very noticeable neck-in of the flow was apparent."

It is interesting to point out that this early concern, with respect to the lack of airflow through the engine, was misleading to say the least, since, in the final arrangement of baffles, the basic idea centered around limiting the volume of air passing the engine to that which passed over the cooling surfaces.

"In an attempt to cool the front bases with the narrow ring cowl in place, special curved baffles were designed and tried on cylinders Nos. 10 and 12. Two positions of the baffles were tested; first, in close to the barrels and then, out about 1.5 in. In both instances the temperatures were raised. Cooling was worse than with no baffles at all."

It should be stated at this point, as found in later tests, that it is quite impossible to interpret correctly the results of baffle tests on only a few cylinders as applied to the results that would have been obtained had these baffles been installed on all cylinders. In some cases the results are entirely contradictory. The reason for this will appear clear in a later part of this paper; but, had this been known during the early stages of development, a great deal of time loss and discouragement would have been avoided.

"Another type of baffle was tried in test No. 23. The baffles were triangular in shape and about 18 in. long. They were fastened to the inner side of the ring cowl and extended from 1 in. behind the leading edge of the ring cowl to 1 in. behind the trailing edge, passing between the ears of the cylinders, both front and rear. They gave an average decrease of 13 deg. fahr., in head temperatures, both front and rear, without affecting the bases. Since the head temperatures were already satisfactory, these baffles were not employed further. . . .

"It was found that an inner flange-cowl, as shown in Fig. 3, lowered the front base-temperature an average of 16 deg. fahr. while increasing the front head-temperatures only slightly. Rear head and base temperatures were a little lower with the flange cowl. In other words, the flange cowl considerably improved temperatures at the hottest points. A speed-course run of July 18, 1931, showed, moreover, that the flange cowl hardly affected the top speed."

Although this inner flange-cowl gave promising results, it was later found possible to abandon it. It is mentioned here to show the extent of the ideas tried in an effort to solve the

problem, and because it is allied with research work now in progress.

"When the N.A.C.A. cowl (full length, 41 in.) was tested, it gave temperatures which were higher than those with the narrow ring cowl. It also showed a gain in maximum speed of 3.2 m.p.h. over that with the narrow ring in the forward position."

Here again higher speeds could be realized but only at the expense of increased cylinder temperatures. Later tests show, however, that high speeds and low temperatures were not necessarily incompatible.

"One more type of baffle was tried as a final attempt to cool the front bases by this means. These curved baffles No. 2 were simply flat scoops bolted to the inner side of the flange cowl. The trailing edges came to about the center line of the cylinders and were pulled in very close to the barrels. They were fitted only to cylinders Nos. 10 and 12. The improvement in cooling with these baffles was too slight to be worth while.

"Finally, on July 24, 1931, a cowling combination was found which was deemed sufficiently satisfactory for running airplane performance, but which still left much to be desired."

At this time, after continued testing from June 23, to July 24, 1931, a satisfactory baffle arrangement had not yet been devised. Upon completion of airplane-performance testing, the baffle-test work was again resumed and continued into December, 1931. Very encouraging results were obtained, leading to the present conception of baffle design. Excerpts from the report on these tests are as follows:

"Twenty-two thermocouples were installed on the heads and bases of all rear-bank cylinders and Nos. 4, 8, 10 and 12 of the front bank for temperature tests Nos. 1 to 49. For the succeeding runs additional thermocouples were installed on the heads and bases of cylinders Nos. 2, 6, and 12, making

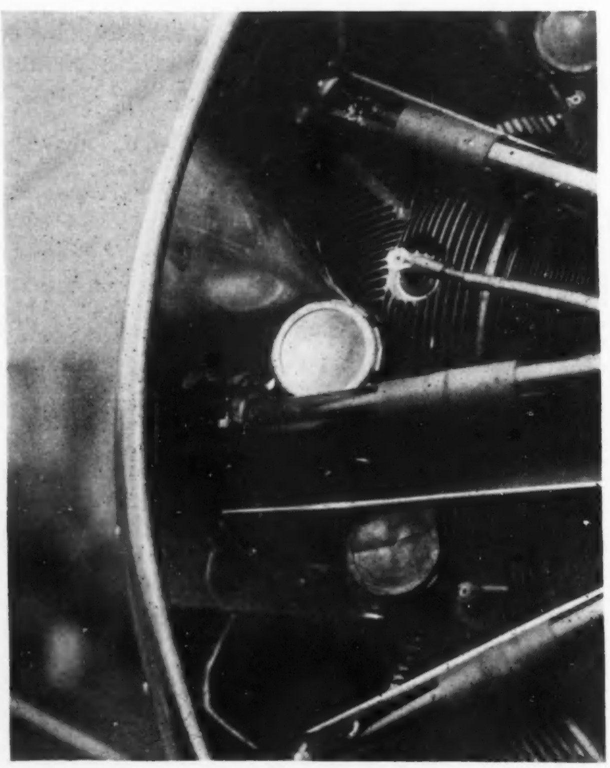


Fig. 5—Engine Equipped with Head Baffles (H_1) and Contour Baffles (C_1)

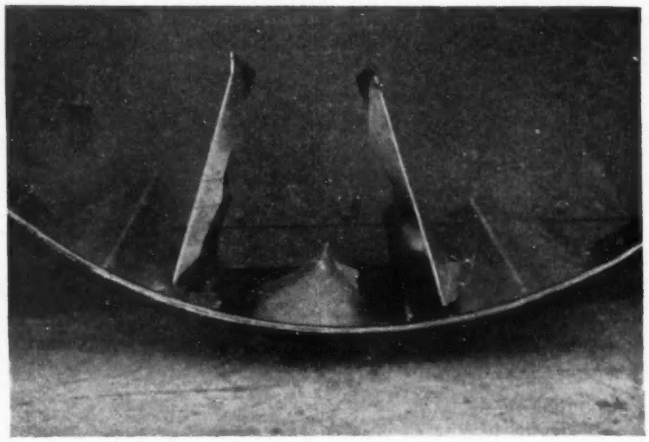


Fig. 6—Attachment of Head Baffles (H_2) and Contour Baffles (C_1) to a Ring Cowl

twenty-eight in all. All cylinder temperatures were measured with a potentiometer.

"Temperatures were measured in both level flight and climb and, unless otherwise noted, with the mixture control set at full rich for the sake of uniformity. The general procedure was to fly level at 2000 ft. with full throttle until the temperatures became steady; then to pull up into a climb at approximately best climbing speed and continue until the temperatures began to drop. All temperature runs were made with the airplane at about 4100 lb. gross weight and with the engine turning very close to its rated 2000 r.p.m. at sea level.

"This series of cowling and cooling tests was begun on Oct. 14, 1931, immediately after the conclusion of the performance tests at two gross weights. All of the first twenty-one temperature tests were devoted to the determination of the best type of flange cowl to be used with the new design of S baffles, (S_2) in the first instance and with the V baffles (V_1) in the second. In addition, a comparison of the merits of the S and the V baffles was sought. See Fig. 4.

"With all three types of flange cowl (Fig. 3), the S_2 baffles showed themselves definitely better for cooling than the V_1 baffles, except for the rear-bank heads; the temperatures of the latter were not critical. This improvement was not a matter of a few degrees, but an appreciable amount. The most marked improvement came in the front bases (average drop of 20 deg. fahr.) and the least in the rear bases (average drop of 1 deg. fahr.). The front heads averaged 9 deg. cooler and the rear heads 24 deg. fahr. hotter with the S_2 than with the V_1 baffles. . . . .

"One other conclusion may be drawn from these two runs. Without any baffles front or rear, that is, with presumably the condition of maximum-volume flow of air through the engine, the front bases are still too hot, the highest temperature being 301 deg. fahr. in full-rich climb. Thus it is evident that any scheme such as cutting openings in the noses of the V_1 baffles will not increase the flow of air enough to attain satisfactory cooling. Apparently when outer cowling is used with this engine, baffles of some sort on the front cylinders are a necessity, regardless of the presence or absence of rear cylinder baffles. . . .

"Up to that time, no S_2 baffles had been installed between cylinders Nos. 7, 8 and 9, because of the oil sump. The sump apparently acted as a sort of baffle itself, for these cylinders seldom showed critical temperatures. The omission of the S_2

baffles between cylinders Nos. 2 and 3, 13 and 14, due to the presence of the front ignition-wire conduits, however, was more serious. These cylinders, particularly No. 3, very often gave maximum temperatures. For temperature tests Nos. 26 and 27, as well as all succeeding tests employing this type of baffle, S_2 baffles were installed between cylinders Nos. 2 and 3, 13 and 14, the lower parts being cut away to make room for the ignition conduits. In spite of the makeshift construction, the baffles lowered the base temperatures of these cylinders an average of about 5 deg. fahr.

"The question arose as to just what was the quantitative effect on cylinder temperatures from adjusting the mixture control for best power in climb. For this reason three climbs were made in succession during the same flight with the conditions for all three identical except the mixture control. The first climb was made full rich, the second with a moderate use of the mixture and the third at best-power mixture. The results indicate that the change in mixture from full rich to best power in climb raises the head temperatures about 45 deg. and the flange temperatures about 15 deg. fahr. As might be expected, the moderate use of the mixture gave increases somewhat less than these. . . .

"In order to determine the net beneficial effect from the best inner cowling and baffling combination thus far found, that is, straight-nose flange-cowl and S_2 baffles, X-26 was flown with no inner cowling. The improvement in cooling (due to the flange cowl and S_2 baffles) amounted to approximately 17 deg. in the head temperatures and 12 deg. fahr. in the flange temperatures. These were average values. . . .

"With the conclusion of run No. 47 no further attempt was made to develop the flange cowl. There is no doubt that it improved the cooling of the flanges to a certain extent, but the gain was not sufficient to lower the temperatures to satisfactory limits. Added to that were the maintenance difficulties which such a cowl would present in service. The complete removal of the flange cowl required the removal

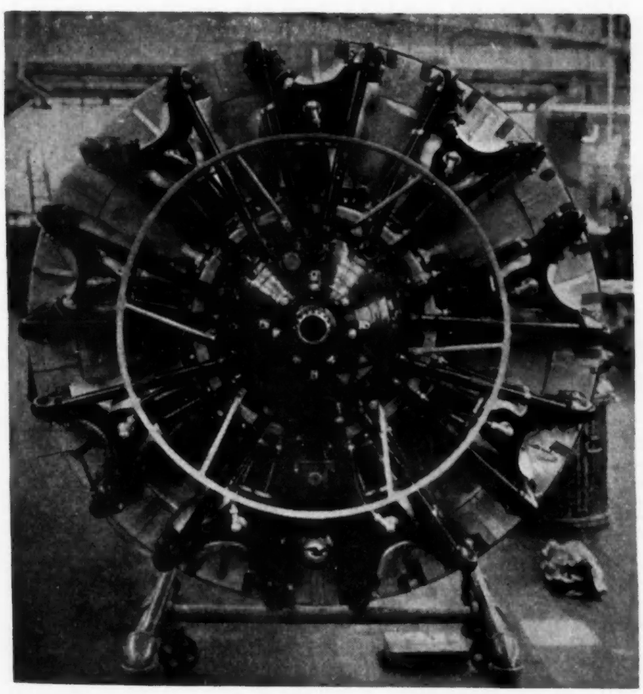


Fig. 7—Front View of Experimental Pressure Baffles on a "Wasp" Engine, Being the Original Baffles before Trimming

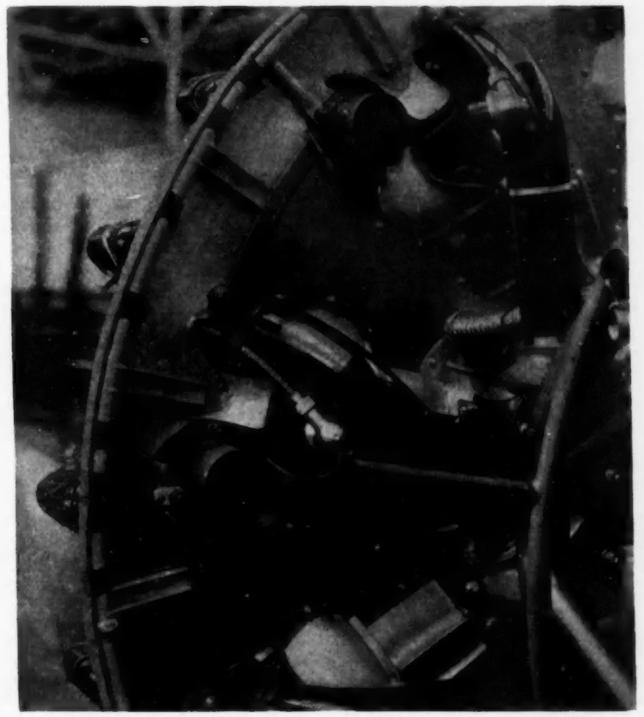


Fig. 8—Rear view of Experimental Pressure Baffles on a "Wasp" Engine, Being the Original Baffles before Trimming

of all push rods both front and rear, with the subsequent checking of all valve clearances.

"Up to this point every effort had been aimed at inducing a large volume of air to flow through the engine disc, while directing its flow as nearly as possible to the points needing cooling. Volume of air, however, was the essential objective. With the invention of what may be termed the plug type of baffle, as characterized by the head and contour baffles, *the objective became the definite limiting of the flow of air through the engine disc, together with an accurate control of the air which did flow, directing it to and over those parts of the engine which most required cooling.*"

This development proved to be the turning point in the battle. From this point on the work was to a large extent a matter of detail refinement and improvement in baffle shape, size and arrangement. As indicated, however, in the preceding paragraphs, an unmeasurable amount of time and effort had been expended in arriving at what may appear to be an obvious conclusion.

Continuing with excerpts from the report:

"The first use of this principle in flight came in tests Nos. 48 and 49 on Nov. 11, 1931. The head baffles employed were in two parts. The first (H_2) almost completely filled the spaces between the ears of both the front and rear cylinders, following closely the outline of the ears. The second part of the baffles (H_{2a}) supplemented the first by blocking, in a similar way, the openings between the front and rear cylinder ears, that is, between two exhaust ears on one side and two intake ears on the other. (Figs. 5 and 6.)

"These head baffles gave most gratifying results, lowering the head temperatures by an average of 56 deg. and the bases 9 deg. fahr. The basis for comparison was a set-up similar except for the use of the straight-nose flange-cowl.

"Continuing the application of this principle, contour baffles (C_1) were designed which replaced the small baffles (H_{2a})

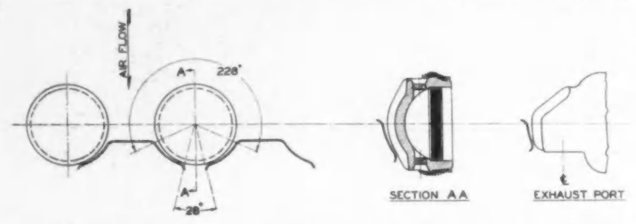


Fig. 9—Typical Arrangement and Location of Pressure Baffles for a Single-Row Engine

between the ears. They extended from the head baffles H_2 down to the cylinder flanges and followed the contours of the cylinders very closely, leaving only small openings next to the fins and, in fact, touching them in many places. These baffles gave the most encouraging results thus far obtained. The full-rich tests first made with them were repeated, using a best-power adjustment of the mixture. In climb, the hottest rear flange was 303 deg. and the hottest front flange 291 deg. fahr. This set-up gave 180.8 m.p.h. over the speed-course, indicating that the head and contour baffles have little if any tendency to reduce the top speed. Contour, head and V baffles were used in all of the succeeding tests."

The real advantage gained through closing off all air passages in the baffle system, other than those required for directing air to the cooling fins, is strikingly brought out by cooling tests on another installation. The baffles were of the new close-fitting type except that during manufacture and installation a few "leak holes" were still present. These were temporarily stuffed with steel wool and the temperatures of the heads were reduced an average of 17 deg. and of the bases 7 deg. fahr.

In order to identify the improved system of baffling where all the space under the cowl was tightly closed off except the small amount required to admit and direct air to the fins, a new name was coined and henceforth these baffles were called "pressure baffles."

To date, the pressure baffles used in the experimental tests were made for the most part from aluminum sheet and the use of tin snips. They were secured in place, both to the engine and the N.A.C.A. cowl, with nuts and bolts and with little time for thought as to ready removability and maintenance. An order was released for a set of factory-built baffles and considerable time and study were given to reduction of weight, simplification of design and ready accessibility to the engine. Careful attention was also paid to the reduction of leaks past the engine except where actual cooling was required. In this new arrangement a sheet-metal diaphragm, to which the baffles were secured, was mounted on a circular hoop of steel tubing, the latter being attached to the engine. This arrangement obviated the necessity for attaching any baffles to the N.A.C.A. cowl and also acted as a mounting for the cowl in addition to providing an air seal between the cowl and baffles.

Tests of this new baffle arrangement on the two-row engine, where special attention was paid to unnecessary leakage of air as compared with baffles similar to the earlier set-up, showed an improvement in cooling as follows: Average reduction of head temperatures, 15 deg. and average reduction of base temperatures, 8 deg. fahr.

Although we had arrived at the pressure-baffle idea, it was still necessary further to refine the design of the parts that directed the flow over the fins of the barrel and heads. Should they enclose almost all or only part of the cylinders? How large an inlet and exit opening should be provided? What should be the spacing between baffles and fin? In order to arrive at results which would be representative of the actual engine installation in the airplane and at the same time reduce the complexity of the problem as applied to the two-row installation, it was decided to carry on these tests on a single-row installation. Since the problem primarily involved the cooling of each individual cylinder, the results should be applicable to any cylinder arrangement.

To this end an experimental set of pressure baffles was designed and constructed for installation on a Wasp single-row engine. The attachment of the baffles was such as to permit ready removal and change of each part in arriving at the most effective set-up. Figs. 7 and 8 show the diaphragm and baffles on the original installation. The cylinders are almost completely enclosed; in fact, there was some doubt in our minds as to whether the engine could be safely flown with this arrangement; but, in an endeavor to at least bracket the results, this extreme condition was used initially, which permitted trimming the baffles on subsequent tests.

The tests, twenty-nine in number, consisted of complete temperature and speed runs. The baffles, head, barrel and ear, were trimmed separately, increasing first the front and then the rear openings; and, lastly, a new set was constructed to check the most effective arrangement found. The tests took place during December, 1932, and January, 1933, on the Vought V-50 airplane. All runs were made at a 2000-ft. density-altitude, at a constant engine power of 550 b.hp., and at a constant engine speed of 2100 r.p.m. The method used in maintaining these constants is as outlined in a paper entitled "Commercial Flight Tests Improved by New Equipment and Methods," by A. L. MacClain and D. S. Hersey[2].

As a result of these tests, the most effective cooling arrangement was found to be as follows:

(1) The clearance space between baffles and fins should be as small as practical. A 1/8-in. space corresponds very nearly to the best results obtained and prevents chafing between baffles and fins.

(2) The head and the barrel baffles should surround about 104 deg. of the head with a front opening of 228 deg. and a rear opening of 28 deg. The opening at the rear spark-plug should be increased accordingly.

(3) The ear baffle on the top of the heads between rocker boxes should cover all fins transversely and, in the fore-and-aft direction, begin at a point about 1/2 in. behind the transverse centerline, extending aft about 2 1/2 in. Allowance should be made to provide an exit opening for the fins around the ports; for example, the baffles between ports can extend somewhat farther aft.

Fig. 9 shows the arrangement of baffles, as described above. Tests conducted later show an improvement if the clearance space between baffles and fin is tapered; that is, if we provide an opening at the leading edge of the baffle of 1/4 in. to 3/8 in., gradually reduced to about 1/8 in. or less, at a point about two-thirds of the way aft from the leading edge of the baffle. The improved spacing only is shown in Fig. 9.

Perhaps the most surprising result of the tests is the size arrived at for the front opening, in that the baffles begin at a point *aft* of the transverse centerline. One apparent reason for this, however, is brought out in the results of airflow tests[3] on a finned cylinder where it is clearly shown that the

[2] See S.A.E. Journal, July, 1933, p. 245.
[3] See N.A.C.A. Technical Note No. 429, Aug. 1, 1932: Heat Dissipation from a Finned Cylinder at Different Fin-Plane to Airstream Angles.

breakdown of flow around the cylinder begins at a point slightly aft of the transverse centerline. The forward finned walls of the cylinder no doubt provide a natural flow path for the air with the necessary high surface-velocity.

It was also gratifying to find that changes from the poorest (initial arrangement) to the final most effective arrangement, although showing a material reduction in temperatures, showed no apparent loss in speed. These results are:

	Average Head Temperatures, Deg. Fahr.	Average Base Temperatures, Deg. Fahr.	True Air-Speed, M.P.H.
First Arrangement	484	270	178
Final Arrangement	416	248	178
Temperature Reduction	68	22	

With all baffles removed, but with the N.A.C.A. and other cowling in place, the average temperatures increased 62 deg. on the heads and 52 deg. fahr. on the bases, and the true air-speed dropped to 167 m.p.h. All of the above temperatures have been corrected to a constant arbitrary strut temperature of 40 deg. fahr. which represented the average of flight tests, by a method checked in many previous flights.

The information gained through the tests on this experimental set-up permitted us to proceed to the design of a practical set of pressure baffles for the single-row engine. The first set was bumped out and fitted by hand, emphasis being placed, however, on ease of manufacture in production and ready removal and replacement. Modifications were made in the engine design to accommodate attachment of the baffles. In order to assure effective operation in service before release for manufacture, these baffles were flight tested and certain improvements made.

The following description covers the results of flight tests on such a set of baffles designed for use and tested on a 3:2 geared 1690-D Hornet in the Vought V-50 airplane. All runs were made at a 3000-ft. density-altitude, at a constant engine power of 700 b.hp. and at a constant engine speed of 2150 r.p.m. The mixture control was set for best power at both level flight and climb. All engine temperatures were corrected to an arbitrary strut temperature of 60 deg. fahr., because it was near the average for this series of tests.

Twenty-eight separate tests were conducted, but those of striking interest were on the engine as tested bare with no N.A.C.A. cowl or baffles, as tested with N.A.C.A. cowling only, and as tested with N.A.C.A. cowling and the best pressure-baffles in place. The following temperatures and speeds were obtained in level flight at best power:

	Average Head Temperatures, Deg. Fahr.	Average Base Temperatures, Deg. Fahr.	True Air-Speed, M.P.H.
Bare Engine Only	420	254	175
Engine Cowling Only	479	268	190
Engine Cowling and Baffles	406	239	191

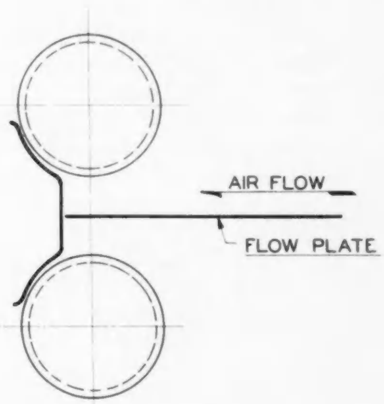
In climb at best power, the following results were obtained:

	Average Head Temperatures, Deg. Fahr.	Average Base Temperatures, Deg. Fahr.	True Air-Speed, M.P.H.
Bare Engine Only	430	262	110
Engine Cowling Only	Too hot—could not climb		
Engine Cowling and Baffles	436	250	110

It has been generally understood that the bare engine alone will always cool better than with some type of speed cowling. Contrary to this belief, the engine, when equipped with the N.A.C.A. cowl and pressure baffles, showed that the speed was not only increased 15 m.p.h., but the cylinder temperatures were also reduced an average of 14 deg. fahr. In the climb condition, the temperature change also slightly favors the pressure baffles. With pressure baffles omitted under the N.A.C.A. cowl, the temperatures in level flight were 73 deg. and 29 deg. fahr. higher for heads and bases respectively, as compared with the pressure baffles in place, and in climb the omission of the baffles gave temperatures too high to continue testing.

With the N.A.C.A. cowling alone, the speed was 1 m.p.h. lower than with the pressure baffles included. The N.A.C.A. cowl used on these tests was, as shown by later data, better suited for use without pressure baffles. Had this cowl been designed to embody an arrangement best suited to the use of pressure baffles, the high speed with the latter, based on data available, would have been increased about 2 m.p.h. Had the cowl at the same time been changed to reduce the velocity of flow through the pressure baffles, thereby raising the cylinder temperatures to agree with the higher temperatures obtained without baffles, the resultant reduction in drag probably would have raised the high speed 5 to 6 m.p.h., thereby increasing the top speed of the pressure-baffle installation from 191 to 198 m.p.h.

On three other installations, where speed comparisons were obtained between the conventional *V* baffle and the new pressure-baffle installation, both equipped with an N.A.C.A. cowl of good design, the increase in speed with the pressure baffles was in all cases about 4 per cent. Emphasis should be placed,



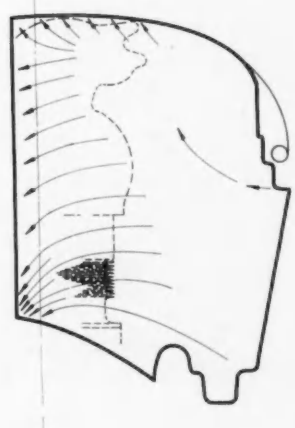


Fig. 10 — Flow Paths of Air Entering Pressure Baffles

The high flow indicated in the lower left corner is due to leakage through a hole in the baffle.

Fig. 11—Rear View of Late-Type Pressure Baffles Installed on a Single-Row Radial Engine

however, on the fact that this increase in speed is not due to the use of pressure baffles in themselves, but rather to the latent possibilities of drag reduction made available through the use of pressure baffles. In other words, the actual drag-reduction results from the cumulative effect of being able to cool with lower volumes of air and hence being able to design more efficient cowls.

Interesting results were obtained, in conjunction with these tests, in obtaining flow paths of the air as it passed from the front cowl opening back to the baffles. Fig. 10 shows one of these patterns, which were obtained by painting flat aluminum plates with lampblack and kerosene immediately before the flight test. The flow lines show that the path of the cooling air is fairly smooth and that its velocity is very low, except at the rear edge of the plate where the air enters the spaces between fins. The indicated high velocity at the bottom point of the baffle is due to a leakage hole between the baffle and the crankcase.

Figs. 11 to 14 show some of the latest developments in production types of pressure baffles for use on single-row and two-row engines. These are now furnished as standard equipment on all two-row models and as optional equipment on many of the single-row engines manufactured by the Pratt & Whitney Aircraft Co.

Engine Cowling

During the progress of the development of the pressure baffles, it was clearly indicated that the type of cowling used had a marked influence on both engine cooling and airplane performance. Before proceeding with the discussion of the cowling development itself, let us consider for a moment the status of the problem at this point in its development. From the data available on the results of the hundreds of tests conducted on the many variations and combinations of pressure baffles, we are in a fair position to believe that we have reached a point of efficiency where a marked improvement through basic changes in the baffle design will be difficult. This statement is based on the fact that the baffles as finally developed are basically sound in principle. They block off all airflow passing the engine except that used for cooling. They direct this air over the hot fin-surfaces.

The clearance between fin and baffle, the length of the baffle, the size of the front opening, the size of the rear opening and the shape of the leading and trailing edges of the baffles have all been explored and the most effective arrangement finally adopted. This involved no compromise between drag and cooling, since, after arriving at a certain point in the baffle design, further changes to improve cooling resulted in no measurable change in performance. It is, of course, obvious that a certain velocity of airflow over the fins is required to cool them properly, but the function of the baffles is not to supply an airflow to themselves, but rather to use the flow supplied as effectively as possible in cooling the engine. In other words, the baffles are entirely ineffective with no airflow but, for any *given* velocity of flow through them, regardless of how supplied, either by the forward motion of the airplane or by means of a pressure blower in front or a suction fan in the rear of the baffles, their effectiveness in cooling the cylinders still remains the same.

Further, and again regardless of how the airflow is supplied, the only power loss from the baffles themselves is in the drag that is produced by the air at this same velocity as it passes through the baffles and over the fins. It is possible and even probable that further developments in air-cooling technique will discover minor advantages to be gained from leading the air to and from the pressure baffles in specially shaped ducts; but these refinements are subsidiary to the basic principle already established.

The problem now is to supply an airflow to the baffles such that the velocity of flow over the fins is just sufficient to keep the cylinder temperatures within limiting value and at the same time to devise a means such that, in the process of supplying this airflow, the resulting loss in airplane performance becomes a minimum.

Whether we use a blower or depend entirely on the forward speed of the airplane, the principle of operation is essentially the same; that is, we direct the outside airstream to the front face of the engine, pass it over the fins and exhaust it into the outer airstream. There are various efficient ways, through cowl design, of inducing a much higher air velocity through the fins than that induced through a normal cowl by the forward velocity of the airplane. However, as we continue to increase the velocity over the fins to abnormally high values through extreme cowl design, we may reach a point where the efficiency of the cowl arrangement is so reduced that the loss in airplane performance is excessive.

It appears that a blower system may be advantageously utilized to act as a booster in obtaining the required air velocity over the fins in the event the velocity of the airplane is so low that only an inefficient cowling arrangement will provide this required fin velocity. Nevertheless, even though we provide a blower to act as an auxiliary, it seems apparent that it can still be used most effectively in conjunction with an efficient cowling system designed primarily to make use only of the forward velocity of the airplane.

The purpose of the following discussion is to outline the results of the work that has been carried on in arriving at what is considered to be an efficient type of cowling arrangement. Basically, it consists of the application of the N.A.C.A. cowl to pressure-baffle cooling. The results are from full-scale tests in connection with the baffle development and from wind-tunnel tests on a model.

Fig. 15 shows a diagram of a typical arrangement. The part of the N.A.C.A. cowl forward of the baffle diaphragm is called the "nose," the part aft of the baffle diaphragm is called the "skirt," the air-exit opening is called the "gill," and the rear cowl adjacent to the gill is called the "shoulder," the latter normally being a part of the engine-compartment cowl. It may be noted that this cowl runs down to the engine

mounting-ring and fits tightly around the crankcase. This was originally considered desirable in providing a smooth air-flow to the gill, but was later found necessary to keep the rear engine-compartment cool.

Due to the time and expense that would be required to develop an efficient cowl from flight testing alone, a one-quarter size (13-in. diameter) wind-tunnel model was constructed and over two hundred separate combinations of nose, skirt and shoulder were tested. A special feature was incorporated in the model to permit application of the results to full-scale installations. No attempt was made to obtain data that could be used in predicting quantitative changes in performance. The purpose of the tests was to obtain comparative effects only.

Fig. 16 shows sectional views of the model. The shoulder, baffle, nose-section and skirt parts are detachable so that they may be replaced by different test pieces. The unique feature of the apparatus is the baffle which takes the place of the engine. It is made of wood and is 2 in. thick. Eighteen radial slots open through it, one of which is cut in a brass insert. This slot has a small static-pressure orifice in its side. A screen is used behind the baffle to diffuse the jets of air.

Flight tests and previous wind-tunnel tests had shown that the drag of a cowling was intimately connected with the amount of air passing through the engine. In the case of a cowl which was well designed for a certain condition, increases in flow always caused increases in drag. As was subsequently found out, increases in overall drag due to increases in flow for the same shape of test pieces were in many cases larger than differences in drag due to differences in the shapes of the test pieces at the same flow. The importance of regulating or at least of measuring flow in comparative cowl tests was thus evident.

The apparatus used fulfilled this requirement. The static pressure in the side of the brass insert served as a means of measuring the velocity of air through the slots. This calculated velocity was checked with a separate calibrating apparatus which measured the volume of flow through the baffle at various orifice-pressure readings.

The quantity of air flowing was a function of the number and size of the baffle openings. In order to obtain results comparable to full-scale conditions, it was advisable to have the velocity of the air leaving the gill in full-scale relation to the free-air velocity. Tests on a complete engine made in the laboratories of the engine company gave data on the amount of air which was passing through the engine when adequate cooling was obtained with close baffles. This quantity, corrected by scale factors for model size and free-air speed, enabled us to run the tests at proper air quantities. The drag of each cowling arrangement was thus obtained as a direct function of the amount of cooling air which it permitted to pass through the baffle.

All of the comparisons among cowls are made at one tunnel-air speed and with one baffle opening. The latter requirement is parallel to flight testing comparative cowls with one set of pressure baffles. It is an important one, because it is essential that the changes in measured drags and flows be caused only by changes in cowlings. A change in baffle would obviously cause a difference in flow. It would likewise cause a difference in drag which would not be representative of a particular cowl because part of the drag change would be due to changing the drag *coefficient* of the baffle. When one baffle only is used, however, any change in the drag of the air passing through the baffle is rightly chargeable to the cowling because it is a result solely of the change in air speed through the baffle caused by the cowl.

In the actual test work, we considered the possibility of cowls showing up differently when tested with different *baffle* arrangements, and to take care of this contingency we made a complete series of cowl comparisons with one baffle and tested the same comparative series with a different baffle, and then again with still a third. This was done by varying the number of opening holes in the wind-tunnel baffle. In each case we found that the relative merits of the cowls ranked in the same order with one baffle as they did with the other baffles. This indicated that cowl comparisons made with one baffle opening truly showed the relative merits of the cowls. This is an important point for flight-test work also, as has been found in our testing.

All of the wind-tunnel cowl-analysis is based on the conception of flow through the baffle as an indication of cooling capacity. While it is evident that there is a close connection between the two, it is of interest to discuss the matter briefly. It brings up a subject which is of vital importance to the aircraft industry, but upon which little work has been published. Within the last year a few pioneer papers on this question have appeared; perhaps the best general survey, with applications is a series of articles[1] by D. R. Pye, previously referred to. This subject might be called Air-Cooling Performance-Calculation and might be defined as relating airplane design and operation to the prediction of engine-cylinder temperatures by the application of existing information on aerodynamics and heat dissipation. The basic principles governing the flow of air through orifices like engine fins and baffles are already available, and there remains the problem of coordinating them with the existing knowledge of heat flow. This last is difficult because of the complexity of the applied theory. Nevertheless, certain general theorems may be em-

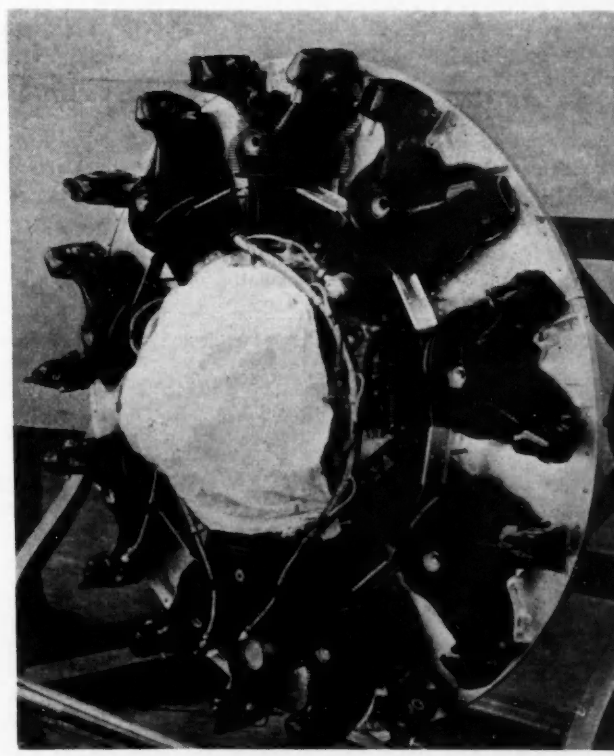


Fig. 12—Front View of Late-Type Pressure Baffles Installed on a Single-Row Radial Engine

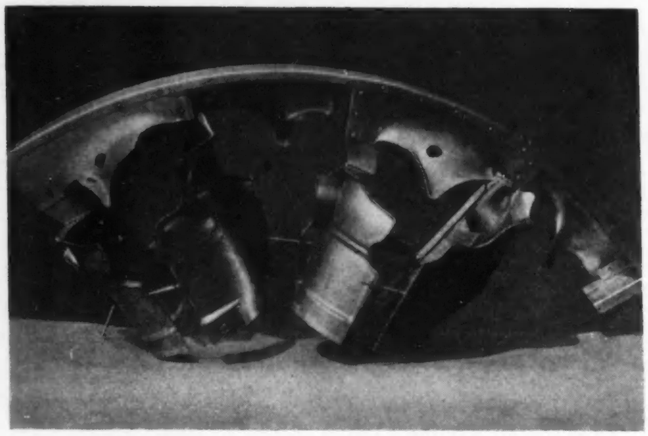


Fig. 13—Rear View of Late-Type Pressure Baffles for Use on a Two-Row Engine

ployed to obtain useful information, bearing in mind that these are subject to a complex inter-relation of minor variables.

The following is an example: The classical equation of heat flow which was derived partly from dimensional reasoning may be rearranged and applied to engine-cylinder temperatures in the following equation:

$$T = k + \frac{k_1 \, b.hp.^{\,x}}{\rho^{\,y} \, V^{\,z}} \qquad (1)$$

where

T = cylinder temperature
k = air temperature
k_1 = a coefficient
$b.hp.$ = engine power
ρ = density of air passing over the cooling surfaces
V = velocity of air passing over the cooling surfaces

The exponent z in Equation (1) ranges from 1 down to much lower values, depending upon where the speed is measured in relation to the fin, the type of flow over the fin, and so forth. The exponents x and y may be taken to be 1 or slightly less. The coefficients k and k_1 depend on a number of things. One group of these relates to the proportion of heat dissipated through the fins and includes overall engine efficiency and heat dissipated through the oil and crankcase. Another group relates to the relation between the measured cylinder temperature and the mean temperature difference between the air and the cooling surface; this is subject to variations in types of fins and baffles, and distribution of heat throughout the engine, such as difference between head and base temperatures. The coefficients and exponents must be chosen carefully. We have used modifications of this equation to estimate *changes* in cylinder temperature due to changes in major variables. In some cases agreement with flight-test data on average cylinder temperatures was remarkably close and, in other cases, large discrepancies have occurred.

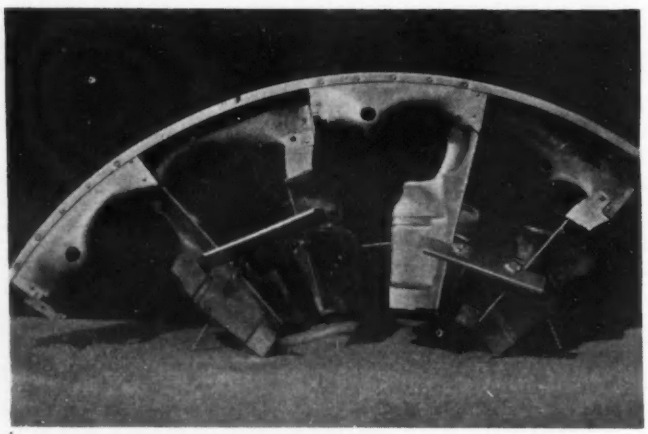


Fig. 14—Front View of Late-Type Pressure Baffles for Use on a Two-Row Engine

This subject is mentioned, not to present the results of research, but to stimulate further work by drawing attention to it and to indicate the influence of airflow through the baffle upon engine temperatures. The equation shows that, except for a factor involving air temperature, cylinder temperatures are inversely proportional to the speed of air through the baffle. We have already given reasons which necessitate running comparative cowl tests at one baffle opening. When this is done, the speed of the air through the baffle is determined only by the quantity of air flowing. Thus, everything else being equal, quantity of airflow through the baffle is an excellent measure of cylinder temperature.

By these methods a wind-tunnel apparatus was made available for obtaining comparative data on many cowl arrangements, cheaply, quickly and accurately.

The first tests conducted were to determine a good shape of nose. Fig. 17 shows the nose shapes tested and the results of these tests. These noses were tested with a good skirt and shoulder combination and the conclusions drawn as to the proper shape should also apply to other skirt and shoulder combinations. Since the diameter of the engine, the diameter of the front air-inlet opening and the location of the propeller control the dimensions of the nose within narrow limits, the main variable is the contour or curvature of the nose. A comparison of the test results shows quite definitely that the contour should be approximately elliptical with the leading edge tangent to a line normal to the thrust line, the trailing edge being tangent to the after skirt-line. This arrangement gave the best results irrespective of the relation between diameter of engine and diameter of front opening; and, conversely, very poor results were obtained with a sharp leading edge, that is, a non-tangent leading edge making a sharp angle with the normal to the thrust line.

Earlier flight tests on nose shapes checked the foregoing. In one case the original nose extended in toward the thrust line, but normal to it. By cutting off the nose at this point—that is, increasing the diameter of the front opening—the true ellipse was still retained and a marked reduction in engine temperature was realized with no reduction in airplane speed. Continued enlargement of the nose opening, which increased the sharpness of the angle of the leading edge, resulted in practically no reduction in engine temperatures, but was accompanied by an appreciable reduction in airplane speed. Making the nose angle still sharper caused some reduction in temperatures, but a very marked reduction in speed. The following tabulation shows some of the results of these flight tests:

Opening, Sq. In.	Shape of Nose	Average Head Temperatures, Deg. Fahr.	Average Base Temperatures, Deg. Fahr.	Velocity, M.P.H.
240	Elliptical	429	264	185
450	Elliptical	391	256	185
650	Sharp Entry	384	255	183
700	Very Sharp Entry	364	250	176

The function of the forward elliptical nose cowl is not only to provide a smooth flow-path for the air passing over the

outer cowl, but also to act as a retainer for the inside air ahead of the baffles. The airflow in front of an engine with baffles, but without nose cowl, would be largely radial. This means that the total pressure of the air at the entrance to the baffles would be much less—and therefore so would the flow through the baffles—than the maximum obtainable, free-air dynamic pressure. Measurements of the static pressure ahead of the baffles with good nose cowls, both in flight tests and in wind-tunnel model-tests, show that the static pressures obtained are from 85 to 95 per cent of the free-air dynamic pressure. The 5 to 15 per cent difference is due to the air velocity in the direction of the fins plus the losses due to the eddying and radial flow which occur even with good nose cowls.

Cowls with excessively large entrance areas are intermediate, in producing flow between good nose cowls and no nose cowls. Wind-tunnel tests of a straight cylindrical nose (Fig. 17) as compared with an elliptical nose, show definitely that the former had a materially higher drag, but at the same time permitted less airflow through the baffles. In fact, the drag with the cylindrical nose was much greater than with no nose at all, and the airflow with no nose was only very slightly less.

The cowl entrance-area can be varied within reasonable limits for a given engine without materially affecting performance. To an uncertain extent, the choice of nose diameter is influenced by the presence of the propeller, an effect not taken into account in the wind-tunnel tests. Although flight measurements with different designs of standard propellers have shown that there is little variation in the static pressure of the air in front of the baffles, it is logical to suppose that the portion of propeller near maximum engine diameter does furnish some increase in the cooling of the heads. Experience with many installations shows that best results are obtained when a larger entrance diameter is used for a two-row engine than for a single-row engine, which has a larger diameter for the same power. This is probably due to a propeller effect rather than to any question of allowing enough air to pass because, in either case, the cowl entrance-area is much greater than the area of the passage through the baffles. In our test work, satisfactory results were obtained when entrance diameters, as a percentage of engine diameters, were within the following ranges: Single-row engines, 65 to 75 per cent, and two-row engines, 75 to 85 per cent.

Raising the percentage above the maximum value for two-row engines has shown a reduction in top speed with little if any improvement in cooling, and lowering the percentage below the minimum value given for single-row engines has increased cylinder temperatures with no gain in top speed. It is probable that further increase in the diameter for single-row engines may be desirable. The opening used should be influenced to some extent by the speed of the airplane in climb and level flight, and for certain ground operating conditions it may be necessary to provide a still larger opening, at some sacrifice in performance, in order to realize sufficient airflow for cooling from the propeller alone.

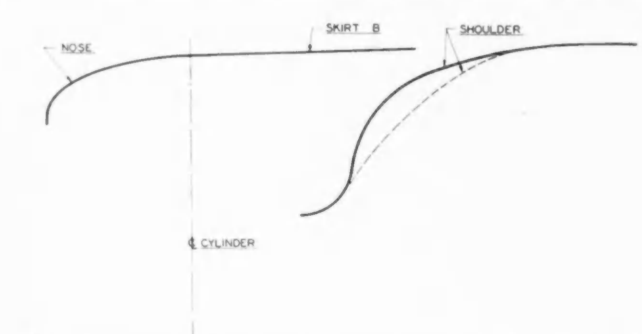


Fig. 15—Typical Engine-Cowling Arrangement

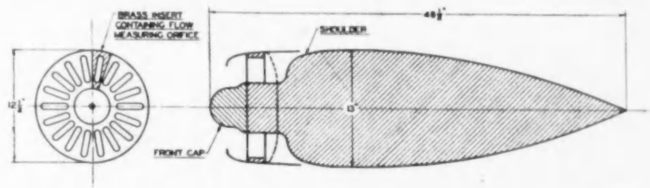


Fig. 16—General Arrangement of Quarter-Size Wind-Tunnel Apparatus

The next step in determining the best shape of N.A.C.A. cowl was investigation of the contour of the "skirt". This problem is made difficult by the fact that the size of the fuselage or nacelle aft of the cowl may be larger or smaller in diameter than the diameter of the engine. Both wind-tunnel and full-scale tests show fairly definitely, however, that least drag is obtained by locating the skirt contour on a faired line extending back to the nacelle or fuselage, as shown in Fig. 18. With a large engine and small fuselage diameter it is difficult to lay in a faired line, but a line extending aft from the skirt should bear into the fuselage line and not extend outside of it. In other words, a small fuselage requires a converging skirt, whereas a large fuselage requires a diverging skirt.

The wind-tunnel tests of a number of skirt combinations having a variation in angle comparable to those shown in Fig. 18 show a maximum variation in drag, for the same airflow through the baffles, of only 5 to 10 per cent. This figure also shows the relative lengths of the three skirts to give in one case equal flows and in the other case equal drags. As in the case of the nose shape, small changes in the angle of the skirt will not materially affect the drag at the same airflow, provided the angle of the skirt does not diverge beyond the faired line extending back to the fuselage. The matter of limiting our conclusions with the remark "at the same airflow," and the means employed in comparing the test results at one airflow, will be made clear later on in the paper.

Intermediate between the design of skirt and shoulder is a question of refined design which may later be found advisable to consider. Should the exit gill of the cowl ever be radially unsymmetrical? On the average, cylinder temperatures in climb are higher for the top cylinders than for the bottom. This, coupled with the fact that many bodies behind the engines are unsymmetrical, indicate that it should. For instance, later test work probably will show that it is best to have a larger exit gap at the top of the gill and at such other places (like the wing of current twin-engined monoplanes) as the air is impeded by obstructions. Still other expedients may be desirable, such as curving the skirt and shoulder in a manner to lead the flow smoothly and evenly around the obstructions.

We have covered the test work in conjunction with the determination of the best contour, nose and skirt, for the N.A.C.A. cowl, and the next step is the determination of an efficient arrangement or shape of shoulder. The real func-

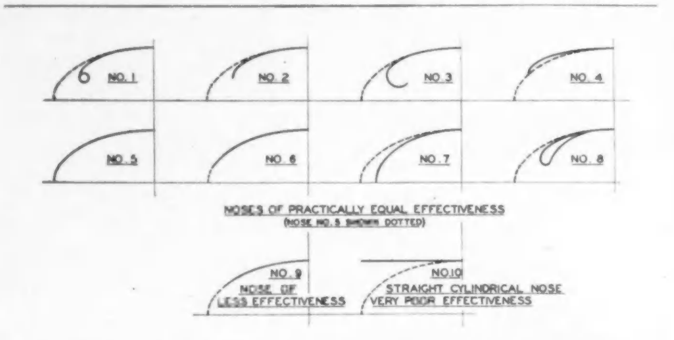


Fig. 17—Cowl "Nose"-Shapes

tion of the shoulder is to provide a flow path for the air leaving the baffles such that, in combining with the external flow over the skirt, the resulting drag is minimum. The only systematic test data available are on the wind-tunnel model; but the general trends shown in various flight tests, and the agreement obtained between flight and model tests on other details of the cowling, indicate that the conclusions arrived at are valid.

The results of the wind-tunnel tests show that the critical point in the shape of the shoulder is near the gill opening. This appears to be entirely reasonable, since it is at this point that the air velocity is materially increased as compared with the velocity in back of the baffles and, further, the curvature of the shoulder at this point combined with the curvature of the skirt, control the direction of flow of the air as it combines with the external flow. Fig. 15 shows two shoulder-arrangements. The difference in drag at the same low flows is negligible, but the shoulder having the easy curvature shows some slight reduction in drag at high airflows. Fig. 19 shows a series of shoulder arrangements similar to those used on certain installations. The test results, as indicated in Fig. 19, again point out that any obstruction in the region of the gill should be as far forward as possible and that the direction of flow through the gill should lead the air smoothly into the external flow.

We wish to emphasize at this point that the designs for a fixed cowl recommended in the original researches by the National Advisory Committee for Aeronautics come remarkably close to those which we conclude to be best for use with pressure baffles.

Controlled Cooling

In the preceding discussion on cowling, all comparisons of the model results have been made on the relative drag of the various arrangements at a given airflow; that is, for comparative purposes at a given cooling effectiveness, at one air speed. No direct information has as yet been presented to show ways of varying the cooling by modifications of the cowling. With a given engine and a given cowl arrangement in a fast and in a slow airplane, the former will show lower cylinder temperatures. How can the cowl be modified to improve the cooling of the slower airplane; that is, increase the velocity of flow through the baffles? The same question may be asked about two engines identical except for power rating.

As pointed out in previous paragraphs, minor changes in a good cowl arrangement may be made without increasing the drag, but the improvement in cooling can hardly be expected to be large. However, it is readily possible to increase the velocity of flow through the baffles by an appreciable amount if we also accept appreciable increases in drag. The first information on the foregoing was obtained through flight tests; and the means employed in increasing the flow through the baffles, thereby reducing the temperatures, was by trimming or reducing the length of the skirt. As the skirt was successively shortened, the cooling improved and the speed dropped off. The average change was on the order of 12 deg. and 6-deg. fahr. drop in head and base temperatures, respectively, for 1 mile loss in speed, except that, as the skirt was successively trimmed, the rate of change in speed became greater for a given change in temperature. In fact, with an originally small gill opening, trimming the skirt 1 in. or 2 in. can be expected to improve the cooling with no noticeable change in speed.

As our work progressed, it became desirable to obtain more data on this subject. Accordingly, several skirts were tested in the wind tunnel at various skirt lengths. Fig. 20 shows curves of model drag plotted against airflow for the skirts shown in Fig. 18. The curves shown were obtained using one tunnel air speed, one nose, one shoulder and one baffle. The flow was increased in each case only by cutting off or shortening the skirts. Polar curves of this sort for two skirts show directly their comparative merits. When one skirt lies to the right of another skirt, it is less efficient because the drag for a given flow is greater than for the other skirt. Sometimes the curves join each other at low flows, indicating that there would be little to choose between the two skirts if they were used on engines that cooled easily or on very fast airplanes. The skirt curves rarely crossed each other.

This method of changing flow was used in determining the relative merit of nose and shoulder shapes; for example, each skirt length of the three skirts was tested with each change in nose or with each change in shoulder. Plotting these results provided a means of comparing noses and of comparing shoulders at any given airflow. In all cases, these curves indicated consistently that the relative merits of the shoulders or of the noses remained the same when tested with any reasonable combination.

The curves of Fig. 20 show that the difference in drag between the skirts is quite small and that the relative merits of the three skirts is unchanged as their lengths are decreased. They also show that the effect of flow upon drag predominates. Tests of other cut-off skirts give similar results. Fig. 21 shows a typical curve of airflow plotted against model drag over the full range for one cowl arrangement having different skirt lengths. This curve is representative of a number that were run.

The general effect of shortening the skirt is quite evident from the curve. When the skirt is so long as to permit no

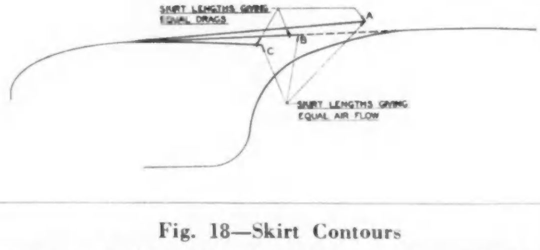


Fig. 18—Skirt Contours

The poorest skirt, becoming worse as the angle is increased is shown at *A*, and the best skirt-contour, fairing into the fuselage, at *B*. The skirt shown as *C* is slightly inferior to that shown as *B*.

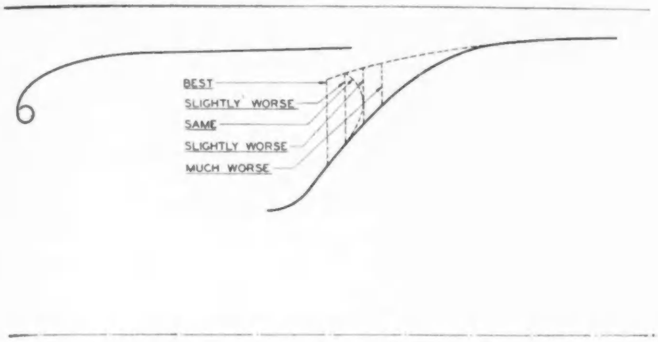


Fig. 19—A Series of Sharp Shoulders

All of these shoulders were less efficient than the continuously smooth shoulders.

air to pass, its drag is still above that of a true streamline body. This fact was established separately. As the skirt is cut off, the airflow passing through the baffle increases rapidly with only slight change in drag. Other skirt-shoulder combinations may yield small decreases in drag for small initial increases in flow. As cutting off the skirt is continued, the flow still increases but the payment in drag has increased still more. The most flow is obtained when there is no skirt at all; in other words, when the ring cowling consists merely of a nose section whose after edge is in the vicinity of the centerline of the engine.

The fact that we believe this increase in drag resulting from an increase in flow in the usual flight range to be typical leads us to an interesting parenthetical conclusion. Five years of general industrial flight-test experience with N.A.C.A.-type cowls have brought forth a maze of contradictory data, resulting in no inconsiderable amount of confusion. This has led to a general suspicion among engineers that there is something mysterious or unpredictable determining the efficiency of engine cowling. Now it may be that the answer is to be found directly in this relationship of flow to drag: General flight testing has not given sufficient weight to cylinder temperatures as a measure of flow and hence of drag. Further research will tell.

Fig. 21 may or may not be correct in indicating relative magnitudes of flow and drag changes. That they are within reason for the flow characteristic is shown by the fact that, using the wind-tunnel data in designing outer cowls, we have arrived at shapes and skirt lengths which coordinate well with our general cumulative experience. It is possible to carry through some interesting performance calculations based on curves of this sort. This must be done with caution for several reasons. The tests included only the effect of a flow-measuring baffle simulating an engine mounted in a nacelle. These differences from full scale are much larger for performance calculations than they are for comparative cowl tests. There is also some question as to scale effect on both drag and flow. Furthermore, any calculation of this type, as the following illustrative one leading to a major conclusion, is based only upon one cowl arrangement modified by cutting off the skirt.

The curve shows that, when the airflow is increased from zero to an amount corresponding to that required for cooling a typical engine, the amount of drag increased. This means that sufficient air to cool the engine caused an increase in the drag of the bare fuselage due to the loss through the engine in combination with the drag at the exit gill. If the engine were assumed to develop the same power in a faster airplane, the flow required from our model—tested at constant air speed—would be less. Therefore, for the faster airplane the increase in drag due to cooling would be less.

For the fast airplane, the skirt length might have to be say 21 in., the other features of the cowling being fixed. The slower airplane, however, would not have satisfactory cooling with this type of cowling if its skirt length were greater than 15 in. The difference in drag between the two cowlings is of the order of 20 per cent. Now in actual practice these two airplanes may be the same; that is, the fast airplane represents the maximum-speed condition of flight, while the slower airplane represents the same airplane in a full-throttle low-pitch climb. This at once suggests the advisability of providing both cowls for the one airplane; in other words, of building a cowl with a variable adjustment which permits a small flow at top speed and much larger flow at climbing speed. The obvious way of doing this would be to design the complete cowling for a cylindrical skirt and simply to slide this skirt fore and aft, increasing or decreasing the size of the exit opening as desired.

This type of variable cooling device would provide improvements of the order of those shown in Fig. 21 and the benefits to be derived from such an arrangement would be considerable. Nevertheless, there were excellent reasons for considering other means of varying the cooling characteristics of ring cowls. One of these was the possible benefits from devices which might be simpler in design application. Another, and the most important reason, was the desirability of increasing the flow beyond that obtainable with a cowl of variable skirt length in order to be sure of providing for the full range of temperatures encountered with variations of weather and airplane operating conditions.

The curve shows that such an arrangement provides considerable improvements in flow through the baffle by means of shortening the skirt. The limit of cooling with such an apparatus is reached when the entire skirt has been removed. This occurs at a flow only about 60 per cent greater than might normally be used. The possibility of even being able to vary the skirt length to obtain this 60-per cent increase in flow is remote because of the mechanical difficulty. It was also desirable to find a means of improving the cooling at a still smaller expense in drag. With these objects in view, other extended tests were run on the wind-tunnel apparatus

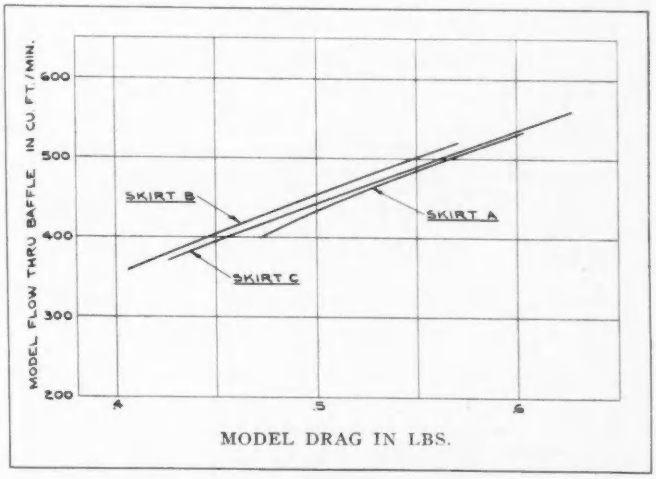


Fig. 20—Curves Showing Drag and Airflow for Varying Lengths of Skirts *A, B* and *C*

used for the fixed cowlings. A large number of different arrangements were tried. Only the more interesting of these will be described.

(1) *Varying Skirt Length.*—This device has just been discussed.

(2) *Skirt Trailing-Edge Flap.*—The idea of the flared skirt or flap originated during early flight tests. It was found that incorporating a fixed flared flap at the trailing edge of the skirt, having a length of about 3 in. and giving an increase in rear skirt diameter of about 2½ in., showed improvements in cooling of 38 deg. and 24 deg. fahr. on the heads and bases respectively, and a loss in speed of 3 m.p.h. The use of this expedient for reducing cylinder temperatures began and led to the development of the final idea of providing regulated temperatures control in the form of a continuous hinged trailing-edge flap, controllable from the cockpit. Wind-tunnel and flight-test results will subsequently be described in detail because this mechanism has proved to be one of the best.

(3) *Skirt Trailing-Edge Split-Flap.*—This term is used to mean a variable flapped skirt, such that the inner surface remains at all times in a fixed position while the outer surface can be set at varying degrees of flare.

The results of tests with a split flap showed, in comparison with the results for the same outer flap alone, about half the flow and about the same amount of drag; that is, the split flap is about half as efficient as the full flap. This may be due in great part to the fact that the exit area of the gill was not increased as the flare was extended.

(4) *Sliding Shoulder-Flap.*—A conical piece was made having the same dimensions as the full trailing-edge flap, but located at the trailing edge of the base skirt. In actual operation it would be nested against the shoulder cowl at cruising speed and displaced forward at climbing speed. This arrangement gave, for a given flap size and angular setting, about the same flow as a trailing-edge flap; but it was considerably less efficient, in that its drag was higher. It is doubtful whether this mechanism could be made to produce as high flows as are possible with the full flap.

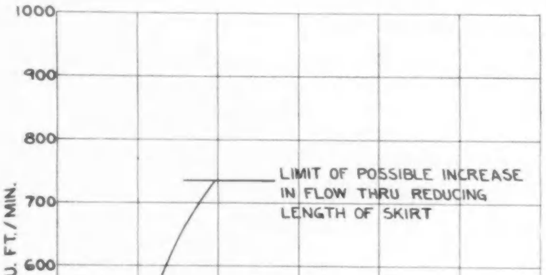


Fig. 21—General Effect of Skirt Length on Flow and Drag

(5) *Trailing-Edge Protuberances.*—During early flight tests on trailing-edge modifications, it was discovered that slight changes on the outer surface of the cowl might produce appreciable improvements in cooling. Mechanisms might be made which would produce such protuberances in flight, as, for example, an inflatable rubber tube. Wind-tunnel tests on a number of protuberances placed on the upper surface of the after edge of the skirt showed that they are about as effective in producing flow, and about as inefficient in producing drag, as were split flaps of the same height. Rounded protuberances are about the same as sharply pointed protuberances of the same height. Very small ones may be well faired so as to be as efficient as full flaps, but they produce only very small increases in flow. The most efficient of those tried formed a kind of venturi at the gill.

(6) *Annular Gill Forward of Trailing Edge.*—Pressure-distribution measurements in flight made over the outer-skirt surface of an ordinary ring cowl had shown that the pressure was much lower near the centerline of the engine than at the trailing edge of the skirt. Therefore, a wind-tunnel test was run on a skirt which had an annular opening just aft of the centerline of the engine to find whether exit area added here was as efficient as area added by cutting off the skirt. The test showed that the forward annular opening improved the flow nearly as much as was obtained by removal of the skirt trailing-edge, but that the drag increase was larger. It was found that a slight improvement in the efficiency of this arrangement could be made by rounding the after edge of this gill, a conclusion previously reached in flight tests on an allied arrangement. Such an opening might be closed by a sliding annular ring. If it were closed by a hinged flap similar to that used at the trailing edge, an improvement might be obtained. More tests disclosed that this type of flap produces about the same increase in drag as would be obtained with the same flap at the trailing edge, but with, however, considerably smaller improvement in flow.

(7) *Separate Openings in the Skirt.*—A number of equally spaced rectangular openings were cut in the skirt. Openings of this sort could be closed by a rotating shutter. Separate openings of this type were found to be less effective in producing flow and considerably less efficient. Such openings might be closed by doors opening in the manner of flaps. This last arrangement is somewhat less effective than trailing-edge flaps and also less efficient. Nevertheless, doors of this type may be made to produce large increases in flow. Tests with varying number and sizes of doors disclosed that, if the doors are placed too far apart, good cooling will be obtained for cylinders immediately in front of the doors but the cooling of the other cylinders will be much less. Therefore, two or three doors are likely to be insufficient and the use of four or more is preferable.

(8) *Variable-Nose Cowls.*—This term is used to designate only devices which modify the shape of a conventional N.A.C.A. ring-cowl nose. Mechanism for actual use of such an apparatus might be by means of rotating shutters or by doors opening in such a manner as to increase the diameter of the entrance without decreasing the length of the nose. Tests showed that these devices offer little promise of being as effective as devices located on the skirt. The air loads at the nose of the cowl, being many times greater than at the skirt, also add difficulties to the development of such a device. As previously stated, more flow is obtained with a

conventional nose than with either a cylindrical nose or with no nose at all.

The use of a conventional nose also results in much less drag. This conclusion was reached after wind-tunnel tests with large flows obtained by means of a trailing-edge flap. Furthermore, in accordance with the processes that produce flow through the baffle, which will be discussed shortly, it is logical to suppose that any change in the cowl shape forward of the baffle can never be made to produce really large improvements in cooling with current skirt designs. This is stated because the most effective variable-cooling cowls obtain their flows by increasing the total pressure drop across the baffle. They have the possibility of lowering the rear pressure by far larger amounts than the nose cowl, which already utilizes nearly all of the maximum pressure (free-air dynamic), can ever raise the front pressure. Variable-nose cowls can be used to keep the engine warm in cold weather, but their use is not apt to result in maximum overall cowling efficiency.

The foregoing analysis shows that our search for the best variable-cooling device was by no means cursory, and it indicates that the arrangement which is most satisfactory from an aerodynamic point of view is the variable trailing-edge flap. Fig. 22 shows the effect of flaps in improving flow and the resultant increase in drag. This curve is typical of flaps just as the preceding curve was typical of the effect of cutting off the skirt. The data for the effect of cutting off skirts are also included on the same curve for the effect of flaps.

The two curves substantially coincide up to the limit of cooling which can be obtained with complete removal of the skirt; and from there on up the flap shows an increasing gain. In other words, a flap which improves the cooling just about as effectively as does varying the skirt length can also be used to obtain much larger increases in airflow through the baffle. These, coupled with the fact that cutting off skirts is as efficient as any known method of improving flow, points unquestionably to the superiority of the flap.

Several different types of trailing-edge flaps were tried as shown in Fig. 23. All of them were modifications of one base skirt. Three different flap-lengths were tried and each of these was tested at various angle settings. Each of the flaps was cut off at least once to represent results obtained with flaps on a base skirt which allowed larger flow. The results obtained were consistent with themselves in all cases. Major conclusions comparing different flaps are as follows:

(1) Hinge point or flap length is a minor variable in determining the efficiency; that is, flow for a given drag.

(2) A short flap turned up to a large angle always produces less flow than a longer flap would at the same angular setting.

(3) Increasing the angular setting of a given flap improves the airflow through the baffle at least up to the range tested; that is, 60 deg. from the base skirt.

(4) Even small flaps may produce large improvements in cooling. The curve in Fig. 22 by no means shows the limit. It is merely the greatest amount of flow that was obtained in the tests actually run, and it results from the use of the smallest of the three flaps. The larger flaps turned up to the same angle would probably produce still larger flows. How far this can be carried before there is a reduction in flow or a prohibitively high drag, we do not yet know.

The procedure of acquiring and using data on cowls is startlingly analogous to current methods of analysis for wing sections. The similarity includes the use of drag, the use of flow as being like lift, the use of cowl shape and attitude as being like wing-camber thickness, angle of attack and so forth. The analogy may even be carried to the point of breaking down the cowl drag into profile and induced drags, cowl-induced drag being a function of flow, of pressure difference and other variables; and the equations for cowl-induced drag are very similar in structure to those for wing-induced drag. This study will not be carried further here because it relates more directly to forced-air-cooling analysis than it does to the design of cowls using only flight-induced flow. Quite possibly, future work on ring cowls may develop masses of data and methods of analysis for them which are as complete and systematic as those which we now employ for wings. But at present both the data and the theory are lacking and therefore ring-cowl design must, for the present, remain on a trial-and-error basis. The best we can do is to give characteristic features of certain cowl shapes and to show what, in general, may be expected of them. This has already been done in this paper, and these remarks show why more detailed instruction cannot now be given for the design of the cowl flaps.

Before discussing the effects of the ring-cowl flaps upon airplane performance and giving flight-test corroboration of the wind-tunnel researches, it may be of interest to mention briefly reasons why the flap so greatly increases the flow of air through the baffle. The analysis of the wind-tunnel test shows that improvement in flow is not a simple function of flap length, hinge point, flap angle, gill exit-area, or increase in diameter of the skirt trailing-edge, or of any simple geometrical variable. The last mentioned comes closest to being a criterion of the amount of flow that will be obtained with different skirts, but use of it underestimates the amount of flow obtained with short steep skirts and overestimates the amount of flow obtained with long skirts set at small angles so as to give the same trailing-edge diameter. This is entirely reasonable, because the flap acts as a deflector for the

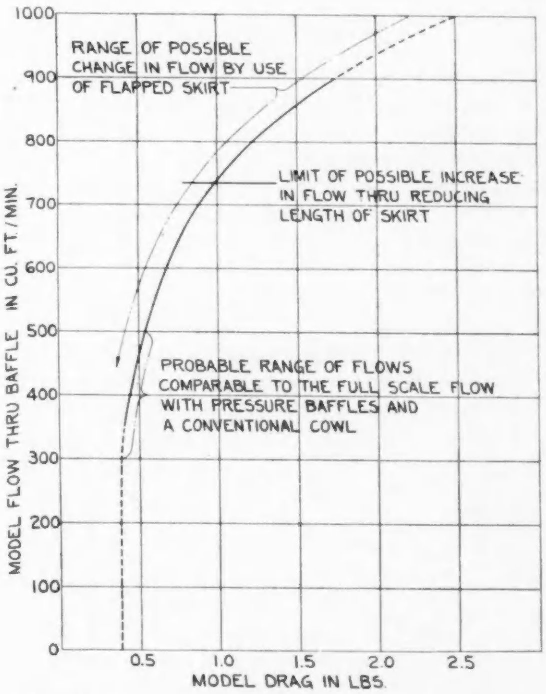


Fig. 22—General Effect of Cowl Flaps on Flow and Drag

air in a manner which is obviously too complicated to express by a single geometric factor. The flap not only acts as a valve in providing a variable exit-area at the gill opening, but also as a means of regulating the direction of flow over the skirt such that flaring reduces the pressure under the trailing edge, which reduction in pressure naturally results in an increase in flow.

As has not been stated previously, the wind-tunnel tests included extensive pressure surveys behind the baffle and in front of it. With the base skirt, a type now used frequently for fixed cowls, the static pressure behind the baffle is considerably above free-air static or atmospheric pressure. The static pressure in front of the baffle is of the order of 90 per cent of the free-air dynamic-pressure and it does not vary much with changes in skirt shape. Thus the front pressure remains high when a flap is used to decrease the pressure behind the baffle. These general conclusions were first observed in flight tests, were then checked in the wind tunnel and rechecked in flight. They are predicated on the assumption that close baffles are used and that all other geometrical features of the cowling and baffling system remain fixed. These observations do not, of course, explain the phenomenon, but merely point out some of its characteristic features.

Considerable space in this paper has been devoted to the acquiring of data on a method for varying the cooling of an aircraft engine, and it has been shown that increased cooling results in increased drag. It has been implied that the use of a variable cooling device will result in improvements in performance and this is unquestionably true. It will now be shown that it is reasonable to expect gains; after that, flight-test data will be adduced to demonstrate that one expectation has been realized.

Some time ago, aircraft engines were cooled very much more easily than they are today. Increases in specific power and the use of propellers which permit the engine to develop full power in climb have made the cooling problem more acute. And yet, aircraft designers have been forced, in their efforts to improve performance, to accept the disadvantages of more difficult cooling while making every effort toward improving the drag characteristics of the airplane.

This increasing difficulty of cooling engines may, if not properly allowed for, result in an underestimate of the value of controllable air-cooling. Naturally, there would be a reduction in expected climbing performance if a flapped cowl were used on an engine that formerly, at a lower power-rating, cooled satisfactorily without it. This is shown by the increases in drag resulting from opening the flap; but, this is not a fair evaluation of the variable cowl unless the resulting improvement in cooling is taken into full account. In other words, the effect on airplane performance of a flapped cowl may be judged solely as an improvement in cooling, allowing for the obvious advantages thereof. This method of evaluation is, however, apt to be confusing and a more direct presentation of the overall improvement can be shown by considering that the flap is used to maintain constant cylinder temperatures under all conditions of flight. This method of comparison is fair and it is almost obvious at the outset that it will show improvements in performance.

Fig. 24 is an illustration of this. For the sake of convenience, the "power-available" curve is drawn for a variable-pitch propeller which permits the engine to operate at constant speed for all full-throttle conditions of flight. In other words, the engine power developed in climb is the same as that at maximum speed in level flight. The "power-required" curve drawn in full is for an airplane equipped with an outer cowl which permits adequate cooling of the engine in climb. As the airplane speed is increased with this same cowling, the heat to be dissipated through the fins remains constant; and yet the amount of air flowing over the fins, and hence, the heat dissipated from them, increases considerably. The cylinder temperatures obtained at maximum speed in level flight are thus lower than those with the same cowl and the same power at climbing speed. If it is assumed that the temperatures in climb are satisfactory, then the temperatures at maximum speed are lower than necessary. This is the same as saying that the drag required to cool the engine is larger than necessary.

Suppose now that the cowl is altered in such a way as to reduce the airflow to an amount just sufficient to cool the engine at maximum speed, but not in climb. The preceding analysis has shown that there will be a reduction in airplane drag if both cowls are reasonably well designed. The next lower power-required curve in Fig. 24 illustrates the change in drag of the airplane due solely to this adjustment of the cowl.

At cruising speed the second cowl furnishes more cooling than is required to maintain the satisfactory cylinder temperatures. This results from the fact that the heat dissipation is reduced roughly directly as the air speed, while the heat to be dissipated is reduced approximately as the cube of the air speed. In other words, the cowl which gave just adequate cooling at maximum speed furnishes an unnecessarily large amount of airflow, and hence drag, for cruising

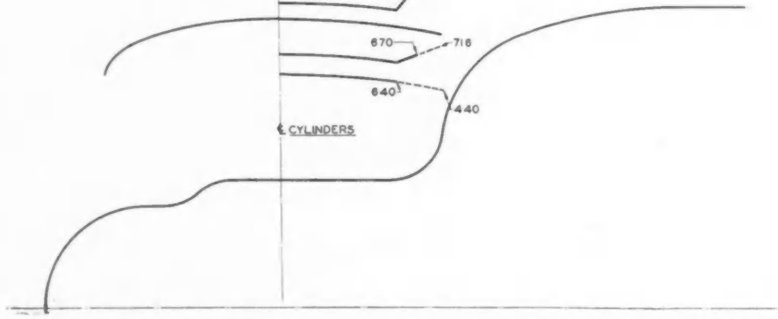


Fig. 23 — Model Skirt-Flap Arrangements Tested

The dotted lines indicate flaps cut to different lengths. The model airflows shown are in cubic feet per minute.

speed. There is thus a further gain to be obtained from designing a third cowl, especially for the cruising condition. The approximate relationship among these three cowls is shown on Fig. 24.

Now these three cowls can be considered as three different positions of an adjustable flapped cowl. While there might be slight improvements in an especially designed fixed cowl over those of a flapped cowl at extreme positions, our test work shows that this is not likely to be the case and that the variable cowl efficiently fulfills the purpose of three or more well-designed fixed-cowls. The uppermost curve in Fig. 24, showing the power required without close baffles and with a fixed cowl set to cool in climb, completes the illustration of the overall gain due to using the entire baffling and cowling system which this paper discloses.

These curves were obtained from calculations based on the use of the equation showing the effect upon cylinder temperatures of the major variables, judicious use of the wind-tunnel data, and cumulative flight-test experience. Systematic calculations of this sort show that the magnitude of the gains in performance which should be expected will be considerably influenced by the type of airplane, by the cooling characteristics of the engine, and by the design of the ring cowl necessary for the base case; therefore, we hesitate to give an estimate of the magnitude of overall gains except in cases where actual test data are available.

Fortunately, even before the data were made available from the wind-tunnel tests, the possibilities of the controllable flapped cowl were sufficiently apparent to warrant the design and construction of such an arrangement for installation on the Vought V-70-A airplane. Figs. 25 and 26 show this installation with the flaps in the full-open and in the closed position. Some flight tests of these flaps have just been completed (Jan. 20, 1934) and the results of these tests are shown in Figs. 27 and 28.

The level-flight tests were conducted at a constant density-altitude above sea level and at a constant engine-power less than rated to eliminate the effect of density upon heat dissipation and airplane drag. The use of a power less than rated, done for convenience of test procedure, resulted in speeds less than the maximum for this airplane. To check the speed effect of opening the cowl, the airplane was flown full throttle over a speed course. These data, shown as the uppermost curve in Fig. 27, verified the previous results.

Fig. 27 shows the changes in cylinder temperatures and changes in airplane speed resulting from changes in the angle of the flap, at, in all cases, constant engine-power. A definite trend in confirmation of expectations and of the model results is apparent. Since we have constant power in all cases, the heat to be dissipated by the fins is also constant. Any changes in cylinder temperatures therefore are due entirely to changes in flow through the baffles. That increasing the angle of the flap definitely increased the flow is apparent from the progressive lowering of cylinder temperatures. This occurred in spite of the fact that the flow due to flaps alone was reduced by a drop in airplane speed. The drop in air speed correspondingly resulted from progressive increases in drag as the flaps were extended. In other words, the flight tests showed a large and continuous increase in airflow through the engine accompanied by some increase in drag as the flaps were extended, exactly as shown in the wind-tunnel tests. The flight tests also show that small deflections of the flap from 0-deg. setting improve cooling with little loss in speed and that the drag increases at a

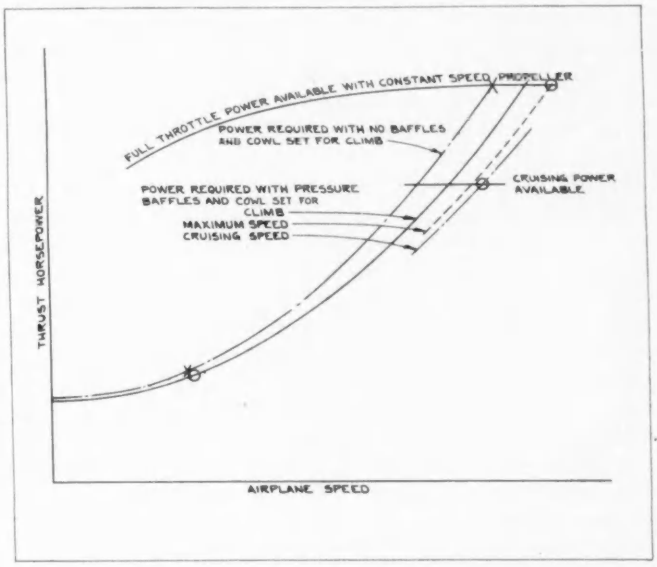


Fig. 24—Overall Effect of a Flapped Cowl on Performance at Constant Altitude and Cylinder Temperature

faster rate as the flap is extended. It is also clearly indicated from the curves that further improvement in cooling at an increased rate (based on flap angle) could have been obtained by further extension of the flaps beyond the 20 deg. used.

An examination of the dimensional changes resulting from extending the flap 20 deg. indicates that, even with flap extended, we have a cowl that should compare quite favorably with fixed cowls obviously designed to take care of a difficult cooling problem. The total cowl length is 32 in., the last 9 in. being the chord of the flap. Extending the flap to 20 deg. increases the radius of the trailing-edge skirt only 3 in. and its angle to the thrust line becomes 15 deg. The gill opening is increased from about 3 in. to 5 in. This last figure is quite reasonable for a fixed-cowl installation which is difficult to cool.

Examining the climb curves, in which the engine powers are comparable to the usual relationship between climb and high-speed conditions of flight, the change in cylinder temperatures between 0-deg. flap and 20-deg. flap-setting is 50 deg. fahr. for heads and 25 deg. fahr. for bases. This incidentally is checked by other flight tests. At the same time, we find that the difference in temperatures at high speed, for 0-deg. flap and 20-deg. flap, is 60 deg. fahr. for heads and 30 deg. fahr. for bases. The advantage, as pointed out in the analytical discussion of substituting a controllable flap on an airplane requiring, for adequate cooling in a climb, a fixed-cowl arrangement corresponding in effectiveness to a 20-deg. flap-setting on the V-70-A, is apparent from these flight tests. With the controllable cowl, the flap should be set at 20 deg. for climb to obtain the required cooling and pulled in to 0 deg. for high speed. The engine temperatures at climb and at high speed will be about the same, and the gain in top speed is, for an airplane whose speeds are comparable to the V-70-A, 6 m.p.h.

The question arises, however, of the effect on rate of climb of extending the flap. The climb data obtained in connection with the temperature readings shown in Fig. 28 have been plotted and the flight-test notes of climb obtained at sea level are: 1720 ft. per min. with a cowl angle of 0 deg., and 1660 ft. per min. with a cowl angle of 19 deg.

The decrease in climb attributed to the flap is only 60 ft. per min., or 3 per cent. The loss in rate at higher altitudes remained about the same, but at 5000 ft. the temperatures began to drop. By gradually pulling in the flap as the altitude is increased, a point will be reached where the flap is at 0-deg. angle and no loss in climb will occur. The airplane ceilings, therefore, will be about the same.

It should be pointed out that, even in the experimental set-up used on the V-70-A, the advantage of the flaps is clearly shown in that this airplane cannot be expected to cool entirely satisfactorily in a full-power climb at 90 deg. fahr. ground temperatures without using the controllable flap. The average head temperatures for 90-deg. strut air-temperature can be expected to run up to 535 deg. fahr. with 0-deg. flap but can be dropped to the very satisfactory value of 485 deg. fahr. by extending the flap to only 20 deg. At full speed in level flight, on the other hand, with flaps closed, the head temperatures at 90-deg. strut will be about 515 deg. fahr. Opening the flap slightly at this high ground-temperature will reduce the temperatures to 500 deg. fahr. with a loss in speed of 1 to 2 m.p.h. Rather than indicating an undesirable cowl arrangement at the 0-deg.-angle setting, it appears on the contrary that, because cruising permits lower flap-setting than maximum speed, the arrangement is really what we are desirous of obtaining; for example, the most efficient basic arrangement (near 0-deg. flap) that will give the required cooling at cruising speed, at high summer air-temperatures. This is based on the assumption that we are primarily interested in the most efficient performance obtainable at the horsepower decided on for cruising speeds.

The foregoing remarks deal for the most part with the possibilities of the use of the controllable flap for different performance conditions. They also point to the equally desirable possibilities of the flap in connection with seasonal air-temperature variations. Not only can the flap be controlled to improve the performance of the airplane during winter in comparison with summer operation, but the temperature of the engine can be adjusted at will to operate, regardless of air temperature, at its most efficient temperature. It will be desirable to lower the cylinder temperatures in summer and to raise them in winter. During recent cross-country flights of the V-70-A, it became second nature for the pilot to regulate the engine temperatures by adjustment of the flap, with entirely satisfactory results. The possibility of being able to control the engine temperatures on a bi-motored airplane during accidental single-engine operation is equally desirable, since slight loss in the speed due to flapping the cowl is unimportant provided the engine continues to function properly.

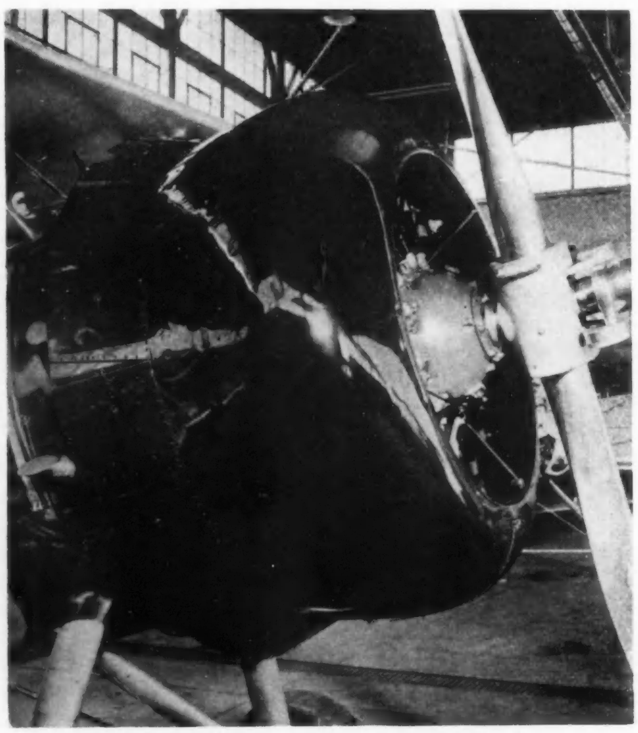


Fig. 25—Open Position of Controllable Cowl-Flaps as Installed on a V-70-A Airplane Equipped with a "Hornet" Engine

It should be made clear at this point that the improvements in performance and other advantages of the controllable flaps, as definitely shown by the curves of Figs. 27 and 28 are, due to the limitations of the tests, merely indicative of the improvements possible and should not in any way be considered as indicative of the limits of improvement. The results are based on a:

(1) Single airplane and a single engine
(2) Single cowl installation
(3) Single length of flap
(4) Small extreme angular displacement of flap
(5) Single gill opening at 0-deg. flap, and
(6) Small variations in air temperatures on a moderately cold day.

With data available on the foregoing items as variables and on the combinations of these variables, the ultimate improvements resulting from the use of the flaps should be greater than the results of these few tests indicate.

In closing the discussion of the controllable cowl, we note the analogy between the use of a retractable radiator on a water-cooled-engine installation, and the controllable-cowl flaps on an air-cooled-engine installation. The functions of the two are identical; that is, to improve the overall performance of the airplane and, at the same time, to regulate engine temperatures according to seasonal variations in strut temperatures.

Other Items of Powerplant Installation.—During the flight tests with pressure baffles, certain changes were found to be desirable and other interesting conclusions were reached.

In the early pressure-baffle installations, the air-intake opening to the carburetor was located aft of the gill, and, although the quantity of air leaving the gill is less with pressure baffles than without, its temperature is correspondingly higher. The normal increase in temperature of the air to the carburetor over the strut temperature was increased, and increases as high as 100 deg. fahr. were observed. To lower the temperature of the air entering the intake, the air duct was led forward through a baffle between adjacent cylinders. The rise in intake-air temperature over the strut temperature was thereby reduced to around 5 deg. fahr. At the same time the air-pressure boost was increased to a value corresponding to that with the best ramming head installations.

It was at first believed that taking air away in front of the baffle diaphragm would affect the cylinder temperature; but, fortunately, test data showed that this was not the case. In other words, there is a difference between air leakage from the front to the rear of the baffle diaphragm and in leading air away from a point ahead of the diaphragm, provided the air does not exhaust into the space aft of the baffles. In the

case of the air intake, the air is led to the carburetor. The effect is very similar to that produced by a slight reduction in the size of the front cowl-opening. The effect of a correspondingly large leak hole, however, is to raise the pressure behind the baffles, thereby reducing the pressure difference between front and rear face of the baffle diaphragm and hence lowering the velocity of the air flowing through the fins.

The method of carrying the air intake through the baffles can be used very effectively on the single-row installations. On the two-row installations, it becomes more difficult due to limited space; but, in any event, if special provision is not made to take in outside cold air *ahead* of the gill opening, the temperature of the intake air probably will be so high as to lower materially the power of the engine. The foregoing remarks also apply directly to oil-cooler installations where it is even more desirable to provide high-velocity cold-air through the oil radiator.

As previously mentioned, it was found desirable to enclose the rear end of the engine and accessories in a tight-fitting cowl to keep the temperatures in this compartment as low as possible. Further improvement in this respect was obtained by leading small tubular air ducts through the baffles back to the rear compartment, with the exit from these tubes directed to the magnetos which, even in the usual conventional installation, may become too warm to operate most effectively.

The use of complete exhaust collectors is becoming more or less standard on all powerplant installations. As previously pointed out, obstructions in or near the gill opening should be avoided. This has a bearing on the location of the collector. It should be located well away from the gill opening, such that the influence on flow is a minimum. There are at least two possible solutions which will accomplish this; one is to locate a collector having a diameter equal to, or greater than, the exhaust-port circle, in the space just to the rear of the exhaust ports. The collector should clear the cowl skirt by a reasonable amount so as not to obstruct the airflow from the rocker-box fins and also to provide cooling for the collector itself. The collector in this location will give the least obstruction to flow from the baffles and through the gill. The foregoing can be used most effectively where the cowl arrangement permits of the use of a long skirt.

The second suggested placement of the collector, which can be used on either single or two-row installations, but is especially adaptable to the latter, consists of housing or burying the collector in the shoulder. The shoulder contour is set up as usual and a D-shaped duct formed by the face of the shoulder and an added circular rear section is provided to house the collector. The individual exhaust stacks lead to the collector proper, through holes provided in the shoulder. The resulting annular space around each individual stack provides entrance for cooling air to the collector proper and, by leading a duct from the housing to the carburetor, an excellent hot-air stove using this air is provided. In arranging the valve or butterfly admitting hot air to the carburetor, means should be provided to permit a liberal flow of air through the housing for cooling when not using hot air.

The use of individual short stacks without a collector ring influences the choice of best cowl-skirt. Some allowance should be made for the reduction in gill opening due to the presence of the stacks, and they should be arranged to give a minimum of aerodynamic disturbance.

While it is not intended to go deeply into the structural design of N.A.C.A. cowls, the following data will be of interest. In conjunction with the tests of pressure baffles on the Vought V-50, with the 3:2-geared 1690-D Hornet engine, pressure-distribution measurements were made across the fore-and-aft length of the N.A.C.A. cowl. These pressures were taken on both the inside and the outside of the cowl. The normal values obtained were plotted on the cowl contour and then integrated to give both the total forward force over the complete cowl and the total force tending to burst the cowl. The tangential forces were, of course, not measured, but in any case they will be of relatively small magnitude.

The results showed higher loads with pressure baffles than without and the magnitudes were surprising. At 167 m.p.h. indicated level-flight speed, the total forward force was 680 lb. and the total hoop tension was 375 lb. These loads were obtained for a cowl of 55-in. outside-diameter and 23½-in. total length. Although the loads given are for the total length of cowl, almost all of the entire load was concentrated ahead of the pressure baffles. The fact that these loads must be given serious consideration in cowl design in the event of high diving-speeds is clearly shown by the fact that the loads at 400 m.p.h. increase to 3900 lb. forward load and 2160 lb. hoop tension. It should also be pointed out that the forward force at local points may well be appreciably greater for a yaw condition.

As previously pointed out, the aerodynamic loads on cowl trailing-edge flaps should be easier to design for than the loads on nose flaps. This is analogous to the relative unit loadings on wing flaps as against wing slats. Although no test figures are available for the loads on controllable-cowl flaps, considerable flight testing with the first one built indicates that they can without doubt be provided for adequately.

The adoption of any new device as a means of improving performance is predicated, in most cases, on the weight involved due to its installation. The weight increase, in the case of the V-70-A airplane, due to the installation of the flaps, control mechanism and supports, was 23 lb. This instal-

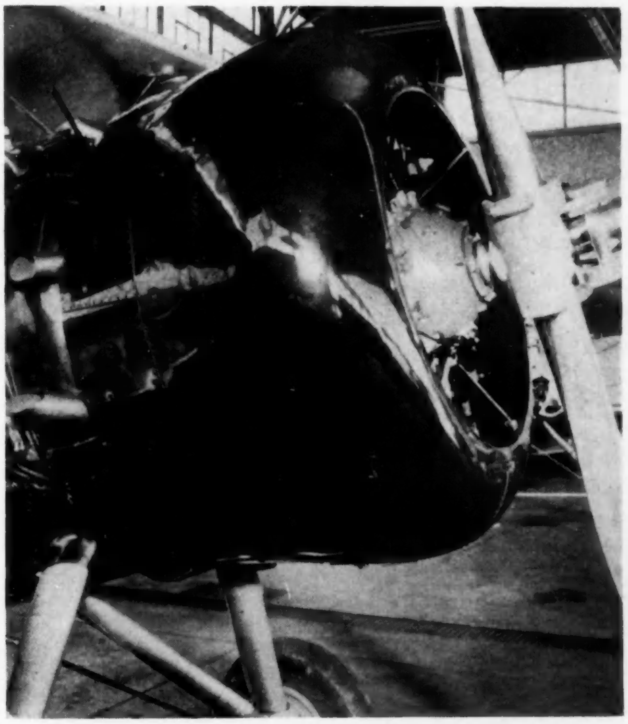


Fig. 26—Closed Position of Controllable Cowl-Flaps as Installed on a V-70-A Airplane Equipped with a "Hornet" Engine

lation, being decidedly experimental and built only to obtain flight-test data, was heavier than need be. Two other designs recently completed show a weight increase due to flaps of only 14 lb. and 16 lb. Considering the device only as being equivalent to an increase in horsepower, we find in the case of the V-70-A that the increase in power necessary to improve the high speed 6 m.p.h. (as the flaps did) will be of the order of 60 b.hp. Even at 1.3 lb. per b.hp. the controllable flap more than pays for itself. However, emphasis should be placed on the fact that, whereas an increase in engine powei not only calls for an increase in weight of the engine, but an increase in weight of structure and other items of the vicious circle, the controllable cowl-flap is entirely a drag-reducing device and the weight increase to the airplane resulting in the use of these flaps is only in the device itself. In addition, the ability to be able to regulate the engine temperature for varying weather conditions is in itself worth some weight increase.

In conclusion, we are glad to acknowledge the friendly interest and encouragement of the United States Navy in this as in so many other United research projects. We would also like to thank the staff of the Aeronautics Department of the Massachusetts Institute of Technology for its valuable work in conducting the wind-tunnel tests.

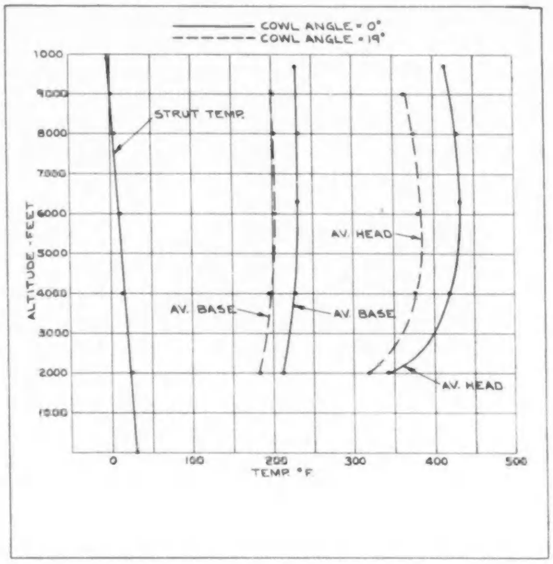


Fig. 28—Results of Climb Tests with a Flapped Cowl at Constant Air Speed with the V-70-A Airplane, with the Propeller Set for the Same Power and Number of Revolutions per Minute in Level Flight at Sea Level

Conclusions

The results of the study show that much can be done to improve the arrangements now in general use by the proper design of engine baffles and cowling. Conclusions as to the type of engine baffling and cowling system which has been found best are given below. These conclusions are advanced as the result of analytical studies, wind-tunnel work and flight testing, and it is probable that operating experience will verify them; nevertheless, they should be accepted as suggestions rather than recommendations.

Engine-Cylinder Baffles

(1) Baffles of the pressure type usually give better cooling than a ring cowl without baffles and they often provide better cooling than is obtained with no baffles and no ring cowl.

(2) These baffles, when used in conjunction with properly designed cowling, produce a considerable improvement in airplane performance.

(3) The function of the baffle is to direct all air which flows through the circumferential circle of the engine so that it passes over the cooling surfaces.

(4) The baffles should direct air to the after quarters of the cylinders only. They need not extend forward of the centerline of the barrels and they may extend aft so far as to leave only a small opening. The rear edges of the baffles should nearly touch the fins and the front should be spaced slightly farther out.

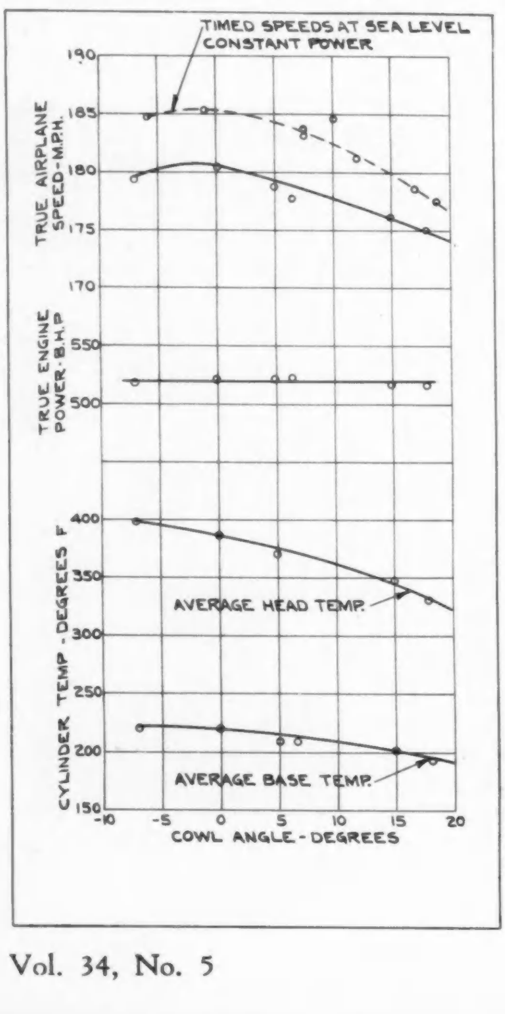


Fig. 27—Results of Flight Tests with a Flapped Cowl in Level Flight at Constant-Density Altitude at 2000 Ft. on the V-70-A Airplane

Engine Cowling

(1) The engine cowling-system which is most suitable for use with pressure baffles is very similar to that originally recommended by the N.A.C.A.

(2) The nose of the ring cowl should have an opening diameter of around 80 per cent of the engine diameter. The nose should have a smooth entry, fairing in to a plane perpendicular to the thrust line. Slight differences in nose shape affect efficiency only slightly.

(3) The after skirt of the ring cowl should be approximately on a line fairing the nose and shoulder cowls smoothly.

(4) The shoulder, wrapper, or accessory cowl should fair into the fuselage in a smooth curve. It should extend quite far forward to provide as smooth an exit as possible.

(5) Variations in cowl shape may, without producing large variations in overall efficiency, cause very large variations in airflow through the baffle with resulting changes in drag. In fact, the amount of cooling air which a given cowl permits to flow through the baffles is one of the most important factors determining its drag. This indicates that (*a*), the cowl should be designed to furnish no more than adequate cooling, and (*b*), previous contradictory test-results may be explainable by this hitherto neglected phenomenon.

Controlled Air-Cooling

(1) Controlled cooling, to provide adequate cooling for variations in air temperature and to reduce engine drag to a minimum for each airplane and engine operating condition, is highly desirable.

(2) A cowl having a skirt composed of a series of separate flaps with a mechanism for flaring them in flight, fulfills this requirement satisfactorily.

(3) A relatively short flap, adjustable through small angles, provided a range of cooling adequate to cope with variations in weather and conditions of flight.

Other Items of Powerplant Installation

(1) The exhaust collector should not interfere with the exit flow and yet should receive sufficient cooling air.

(2) Special provision should be made to prevent overheating of the accessory compartment.

(3) Carburetor intake air in most cases should not be drawn from behind the baffles where the pressure is low and the temperature is high.

Causes and Effects of Sludge Formation in Motor Oils

By D. P. Barnard,[1] E. R. Barnard,[1] T. H. Rogers,[1]
B. H. Shoemaker[1] and R. E. Wilkin[2]

SINCE it appeared reasonably certain that the key to the presence of sludge deposits, regardless of form, lies in the oxidation of the oil to give the small amounts of asphaltic substances which are inevitably present, the authors prepared the following paper to summarize and appraise the evidence on this point and to present a method for measuring oil stability.

After discussing sludge, its composition and effects, engine observations of oil oxidation are stated and the conclusions are reached: (*a*) that appreciable oxidation of each of the oils occurred in a 50-hr. run; that (*b*), variation in engine output markedly influenced oxidation of the oils while variation in sump temperatures did not; and (*c*), that even with the wide difference in the character of the oils used, chloroform insoluble material ("carbon") did not vary greatly but appeared to be determined mainly by engine condition.

A comparison of laboratory and engine tests is made and the Indiana Oxidation Test is described. Service observations are stated and commented upon. In conclusion it is said that the indication is that the formation of sludge in motor oils is due primarily to asphaltenes resulting from oxidation of the oil.

THE term "sludge" has been used for years to denote virtually any deposit—except combustion-chamber carbon—found in an internal-combustion engine. While it is generally understood that sludge deposits originate in the deterioration of the lubricating oil there is no uniformity in the constitution, or even in the appearance, of such accumulations. Analyses have shown compositions varying from quite large proportions of asphaltic material which undoubtedly did come from the oil to almost negligible proportions of such substances, with the major part of the deposits consisting of "blow-by" carbon, inorganic material and, occasionally, water. In appearance, aside from their invariably being black, these deposits differ quite widely, the range extending from a thin hard "lacquer" to a soft putty-like mass of considerable bulk. Frequently, deposits in the form of gelatinous masses are encountered and, in fact, such accumulations are usually responsible for the clogging of screens and filters. In such cases the non-fluid accumulation generally consists largely of oil in which carbon and inorganic material are held in suspension by the presence of some small amount of asphaltic matter. The results arising from the presence of these deposits are almost as varied as their consistencies and range from accumulations in oil pans and other relatively stagnant locations which, when an engine is dismantled, appear quite objectionable but which actually do no harm, to deposits in such vulnerable spots as ring grooves, oil screens and filters where actual interference with engine operation results. It appears reasonably certain that the key to the presence of sludge deposits, regardless of form, lies in the oxidation of the oil to give the small amounts of asphaltic substances which are inevitably present. It is for the purpose of summarizing and appraising the evidence on this point and presenting a method for measuring oil stability that the following paper has been prepared.

Sludge, Its Composition and Effects

It is generally recognized, particularly by those actively engaged in engine maintenance, that sludge may appear in a variety of forms ranging from deposits quite soft and gelatinous in nature to those of the texture of coke. Regardless of their physical characteristics the analyses of sludge deposits show in general very little material which can be positively identified as having resulted from oil deterioration, there always being some doubt as to whether the carbon present results from "blow-by" of products of combustion. Even in the case of deposits occurring on the under side of piston heads, the method of their formation has not been clearly understood. In spite of the fact that sludge deposits may consist either almost entirely of carbon or, at the other extreme, of unchanged oil, a certain consistency has been

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noted in that they are invariably found to contain some asphaltenes which are materials insoluble in petroleum ether (or "hexane") but soluble in chloroform, which material is undoubtedly the product of oil deterioration. For example, the results of analyses of a large number of used crankcase oil and sludge samples yielded the values given in Table 1.

Data of the type given in Table 1 lead at least to the suspicion that, even though most of a sludge may consist of substances other than those arising from oil deterioration, nevertheless such deterioration products are necessary components if only to serve as binders and peptising agents.

Many of the visible effects of sludge formation are quite familiar, particularly the clogging of screens, passages and filters. It is of course unnecessary to dwell upon these troubles in detail, as it is well known that the two former effects may be responsible for engine failure while the third prevents the filter from performing its duty of cleansing the oil of foreign matter. However, Figs. 1 and 2 may be referred to as examples of deposits of this type.

The effects on engine performance and life due to stuck piston-rings, however, do not seem to be quite as well known although, actually, it appears that ring sticking is one of the most important phases of the sludge problem. A few examples may serve to illustrate this fact. In Fig. 3 is shown a piston from a test engine which failed at the end of 37 hr. of a projected 50-hr. run due to stuck rings and subsequent seizure of the overheated pistons. It was necessary to rebuild this engine following the failure, as the pistons were completely ruined. This represents a rather exaggerated case, but it does show the same behavior that is frequently observed in service over longer time periods.

Fig. 4 shows an oil-consumption curve for this same engine in which the average oil consumption for a 50-hr. test-period is plotted against the number of rings found to be stuck and inoperable at the end of that period. Two sets of data are given in this chart covering two different oil sump temperatures, all other operating conditions being held constant. Many other mechanical causes of high oil consumption in engines are well known, but it is reasonable to expect that rings sticking, either permanently or temporarily, may be responsible for many of the observations of high and erratic oil consumption which are frequently reported. Further, not only do sticking rings impair the output of any particular engine, but susceptibility to this trouble very definitely limits the permissible outputs of truly high-specific-output engines such as are necessary in the aviation field.

The two-cycle engine also suffers much from ring trouble when serious attempts at high mean effective pressures are made. In this particular type, of course, the problem is aggravated by the hot gases playing directly on the exposed portions of the upper rings at the moment of exhaust-port opening. That ring sticking can be such an important limiting factor in engine performance is inevitable from the fact that, as rings become stuck, heat transfer becomes impaired with the result that piston temperatures increase. In the aviation engine, stuck rings are liable to result in piston failure and probable wrecking of the engine. Even if actual failure does not occur, overheated pistons tend to induce detonation and, to a certain extent, seizure and abrasion. Occasionally a badly carboned and stuck set of rings will result in greatly exaggerated cylinder wear, which is no doubt accentuated by the fact that under such conditions the filter will also in general be inoperative. No exact valuation can be placed on these difficulties, but it is quite safe to assume that any reasonable means of their elimination would be well justified.

A number of miscellaneous examples of sludge are illustrated in Figs. 5 to 13 inclusive which are given for the purpose of illustrating a part of the variety of forms in which sludge deposits occur and, by way of contrast, the results obtained by operation on an oil not susceptible to sludge-forming oxidation.

The foregoing discussion may be summarized briefly by stating that, although the bulk of most sludge deposits may consist of "blow-by" carbon and other matter foreign to the lubricating oil, the actual existence of a deposit is in general due to the presence of products of deterioration and, probably, oxidation of the oil.

Engine Observations of Oil Oxidation

While it has been recognized to a certain extent[3] that oxidation is an important factor in the service of crankcase oils, this is not always the conclusion[4] nor have the sources of the insolubles commonly reported in used oils been defi-

Table 1—Analyses of Used Oil and Sludge Samples

Material	Used Oil	Sludge
Number of Samples	192	109
Percentage Insoluble in Petroleum Ether		
Maximum	1.59	80.00
Minimum	0.07	0.17
Average	0.47	15.89
Percentage Insoluble in Petroleum Ether but Soluble in Chloroform (Asphaltenes)		
Maximum	0.81	27.3
Minimum	0.02	0.01
Average	0.17	3.09

Fig. 1—Gelatinous Deposits Taken from a Small Automobile Engine

Approximately 3 liters of this material were removed from the engine.

[3] See the *Journal of the Institute of Petroleum Technicians*, vol. 12, p. 582, 1926, Moore and Barrett; see also the *National Petroleum News*, Aug. 13, 1930, p. 63, Lederer and Zublin.

[4] See the paper by Dietrich presented at the Symposium on Motor Lubricants during the March 5, 1933, meeting of the American Society for Testing Materials.

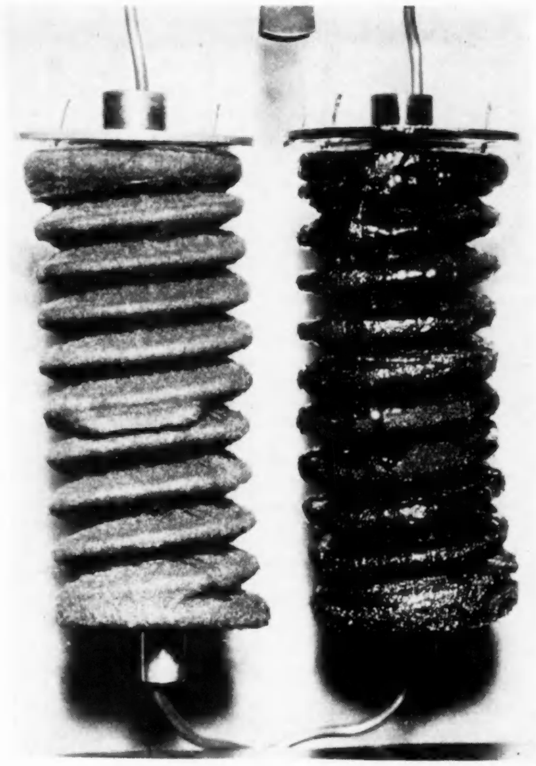


Fig. 2—Clogging of Filters by Sludge
The filter on the right became clogged with sludge. The filter on the left was used with a non-sludging oil, and is still operative.

nitely explored. To establish some basic information on these two questions, a series of exploratory tests were carried out in a water-cooled engine on the following oils:

(1) A "white oil" which oxidizes autocatalytically at 210 deg. fahr. to form colorless acidic oxidation products beginning after about 75 hr. of continuous oxidation. This reaction can be prevented virtually indefinitely by the use of antioxidants. At higher temperatures, 300 deg. fahr. or above, asphaltic materials are formed from the acids.

(2) The "white oil" of item (1) containing an antioxidant not effective above 275 deg. fahr.

(3) A conventional motor oil which, upon oxidation, required a temperature of 330 to 340 deg. fahr. to form appreciable quantities of asphaltic substances in 50 hr.

(4) An unfinished distillate which upon oxidation formed asphaltic materials several times as rapidly as did oil No. 3.

This work may be summarized by the following statements, which represent the conclusions which were drawn at the time this stage of the work had been completed.

(1) Appreciable oxidation of each of the oils occurred in a 50-hr. run. The white oils formed mainly acids, while oils Nos. 3 and 4 formed acids and asphaltenes. Oil No. 4 formed a much greater amount of asphaltenes than did oil No. 3, indicating that sludging is definitely a function of oxidation stability.

(2) Variation in engine output markedly influenced oxidation of the oils, while variation in sump temperatures from 130 to 210 deg. fahr. did not. Oil No. 1 formed oxidation products at a linear rate and oil No. 2 gave identical results with oil No. 1. All of these facts tend to prove that oxidation took place primarily on the under side of the pistons and at other places well above sump temperatures.

(3) Even with the wide difference in the character of oils used, chloroform insoluble material ("carbon") did not vary greatly, but appeared to be determined mainly by engine condition. In the white-oil runs this material appeared in a few minutes, without any accompanying evidence of oxidation of the oil, and it was concluded that the source of such material is combustion-chamber blow-by.

Comparison of Laboratory and Engine Tests

Comparative Results with Various Laboratory Tests.—Having established that oxidation is an important factor in the performance of motor oils and that oils differ markedly in their behavior in this respect, the next consideration was the development of a test method by which the oxidation behavior of oils in service may be predicted. A great deal of attention has been paid to oxidation tests for light lubricating oils such as transformer and turbine oils, all of which involve the use of relatively low temperatures. As it is apparent from the data in the previous section that temperatures well over 300 deg. fahr. are concerned in motor-oil oxidation, none of these tests are suitable on account of the time required, if for no other reason. Tests which have been investigated are as follows:

(1) Fixed-Time Tests at High Temperatures.—These were:

(A) Sligh Oxidation Test[5].—In this test a 10-gram sample sealed in a flask containing oxygen is heated for 2½ hr. at 200 deg. cent., after which the oil is analyzed for naphtha insoluble.

(B) Oxidation in Air at 450 Deg. Fahr.—This is the test described by Davis and Blackwood[6], in which the oil is tested for viscosity change, acidity and naphtha insoluble after 12-hr. oxidation. This test has been used with variable time making it more similar to Tests *C* and *D*.

(2) "Life" Tests at 300-375 Deg. Fahr.—It has been found

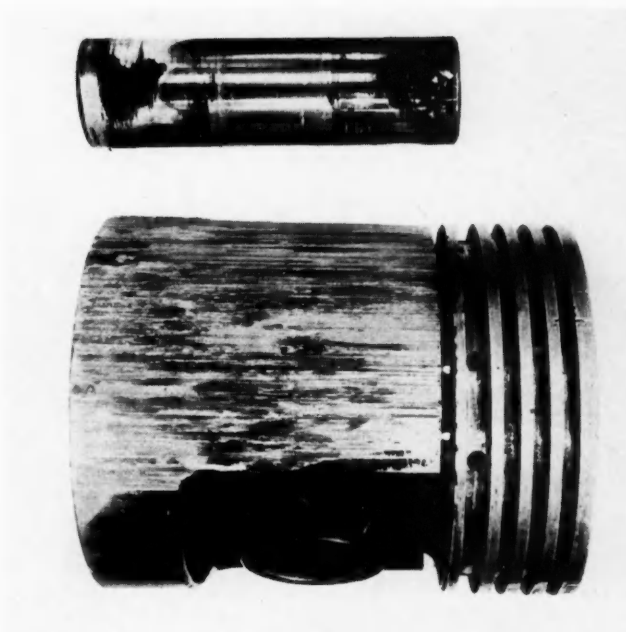


Fig. 3—Piston Failure Due to Ring-Sticking and Subsequent Overheating
The "lacquer-like" deposits on the piston-pin and on the piston-pin bosses should be noted.

[5] See the *Proceedings of the American Society for Testing Materials*, vol. 24, 1924, pp. 964-972.
[6] See *Industrial and Engineering Chemistry*, vol. 23, 1931, p. 1454.

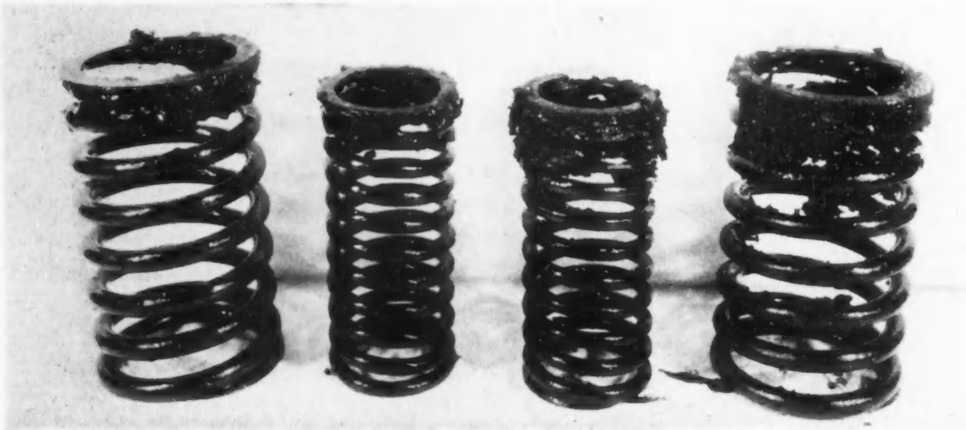
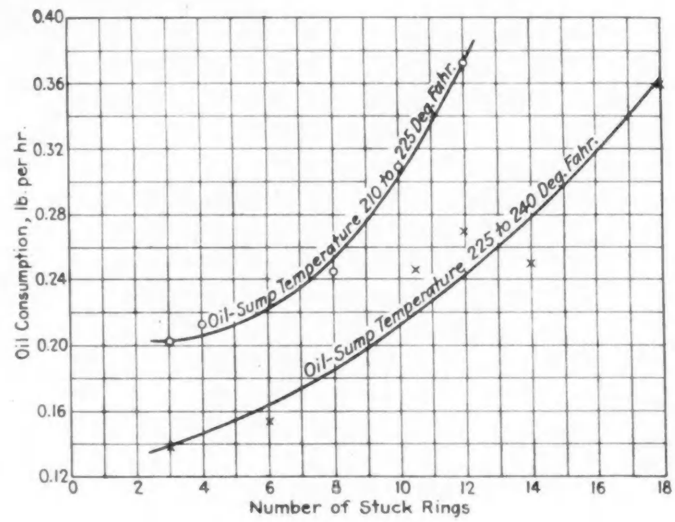


Fig. 5 ***(left)*****—Sticky Deposits on Valve Springs from an Automobile Engine**

Fig. 4 ***(below)*****—Effect of Stuck Piston Rings on Oil Consumption**

The curves show increase in oil consumption with the number of stuck and inoperable rings. The two curves are for different oil-sump temperatures. Other conditions are constant.



that the results obtained by continued oxidation of an oil under fixed conditions of temperature and oxygen supply afford a much more complete and informative picture of its oxidation behavior. In early work some tests were made at 268 deg. fahr. but, as brought out in a previous section, this caused much less asphaltene formation than is experienced in engine operation for a similar time. Most of the work has been carried out at temperatures from 305 to 375 deg. fahr., using either oxygen or air, and two specific tests were extensively used and compared with engine performance.

(C) 305-Deg. Fahr. Test with Oxygen.—In this test oxygen at the rate of 10 liters per hr. was passed through 300 cc. of oil maintained at a temperature of 305 deg. fahr. Samples taken periodically were examined for acidity, viscosity, naphtha insoluble and sometimes other properties.

(D) 341-Deg. Fahr. Test with Air.—This test was carried out in the same general manner as *(C)* except for the use of a different temperature and air instead of oxygen. A detailed description of this test is included in a later section.

All of the above tests, *(A)* to *(D)* are carried out in glass flasks without metal present. With the exception of the Sligh test, provision is made for a supply of oxygen in excess of the rate consumed by the oil. A detailed study of the Sligh test has shown that, with the higher-viscosity oils, which may be characterized as having a higher oxygen absorption, sufficient oxygen is not present.

Test *(B)* gives single values for change in viscosity and acidity, for evaporation loss and for formation of insolubles. Tests *(C)* and *(D)* give curves which show the development of acidity, viscosity and insolubles with time. This is an important feature, as will be seen from an inspection of the curves in Fig. 14, showing some typical tests. During the initial period of oxidation the amount of insolubles is almost negligible. Once the formation of asphaltenes begins in appreciable amount, the rate becomes increasingly greater until a maximum is reached; after this it becomes almost linear. Similarly, viscosity does not change in simple fashion with time. For these reasons fixed-time tests cannot furnish an entirely satisfactory comparison of oxidation stability of oils.

As, from the practical side of the problem, one is chiefly concerned with the service which may be obtained from an oil before sludge formation begins, this initial period, the length of which varies with the oil, is the most significant measure of the sludging behavior of the oil. Accordingly, sludging results obtained in Tests *(C)* and *(D)* are expressed in terms of the time required to form 10 mg. of naphtha insolubles per 10 grams of oil (0.1 per cent). This 10-mg. value, which in a sense is an arbitrary choice, has proved to be very satisfactory as an index because the amounts of insolubles formed during the initial period are usually well below this, the rate generally increasing rapidly when the 10-mg. concentration is reached.

The asphaltene development observed in the engine tests is similar to that of the laboratory test (See Fig. 14), although, before asphaltene formation has become pronounced, there is a somewhat more gradual development. This may readily be accounted for by possible pickup of small amounts of asphaltenes accumulated in inaccessible places which are not thoroughly cleaned after each test, as well as to the fact that only a part of the oil is undergoing oxidation at a given time. For this reason the time required to develop 20 mg. of sludge per 10 grams of oil in the engine has been arbitrarily set as the engine sludging-time. After the oil has reached an asphaltene concentration of 15 to 20 mg., the rate accelerates very rapidly.

Comparison may now be made between the engine tests and the oxidation-test results, by methods *(A)*, *(C)* and *(D)*, as shown in Table 2. It is obvious that the Sligh test reflects in a measure the behavior of an oil in service, but it is also equally obvious that the test is not suited for quantitative correlation. Oil *A* used in runs Nos. 20 and 25 had what is considered a low Sligh value—4.5 mg. per 10 grams of oil—but its service was immensely inferior to several other oils giving Sligh values only a few milligrams less. Similarly, the correlation for the oil *G*, run No. 26, is quantitatively far out of line. With these very definite indications, supplemented

Table 2—Preliminary Engine Tests

Run No.	Oil	Viscosity at 210 Deg. Fahr.	Temperatures, Deg. Fahr.: Sump.	Temperatures, Deg. Fahr.: Average Head	Inspection of Oil After Tests: Oil Consumption Lb.	Inspection of Oil After Tests: Viscosity at 210 Deg. Fahr.	Inspection of Oil After Tests: Acidity, Mg. of KOH per Gram	Inspection of Oil After Tests: Naphtha Insoluble (Mg. per 10 Grams) "Carbon"	Inspection of Oil After Tests: Naphtha Insoluble (Mg. per 10 Grams) Asphaltenes	Engine 20-Mg. Time, Hr. (Asphaltenes)	Oxidation Tests: (A) Sligh	Oxidation Tests: (B) 305 Deg. Fahr.—O_2	Oxidation Tests: (D) 341 Deg. Fahr.—Air
14	*B*	108	230	498	3.4	140	0.99	1.2	0.3	>50	1.0	206 hr.	>350 hr.
15	*C*	123	237	456	2.6	143	0.73	1.6	1.7	>50	0.16	115	>250
16	*D*	114	244	461	2.9	167	....	1.8	0.8	>50	....	...	>250
17	*D*	114	240	491	3.2	163	1.1	4.3	0.7	>50	....	...	>250
18	*C*	123	239	507	3.9	146	2.0	3.3	1.5	>50	0.16	115	>250
19	*E*	158	237	505	3.9	191	0.62	2.4	0.4	>50	1.1	...	>250
20	*A*	90	237	495	5.2	131	0.65	3.5	70.0	?	4.5	31	30
22	*D*	114	235	503	5.6	129	1.8	2.0	0.5	>50	....	...	...
24	*F*	85	239	490	6.8	118	1.1	2.2	0.5	>50	....	32	160
25	*A*	90	239	500	12.0	142	1.7	7.4	274.0	21	4.5	31	30
26[a]	*G*	65	232	525	13.7	94	1.7	11.5	169.0	16	28.2	13	...
27	*H*	102	237	545	8.0	136	0.7	6.3	30.2	44	3.5	36	52
28[a]	*J*	92	239	520	12.8	118	1.7	15.1	92.5	22	3.8	22	20
29	*K*	107	235	520	11.2	126	0.8	7.8	3.2	>50	2.3	80	...

[a] 40-Hr. Runs.

by a large amount of laboratory data indicating its lack of suitability, no further attention was given to this test.

Test *(B)* was studied on a group of four oils of S.A.E.-60 grade, the results of the engine tests being collected in Table 3 (Table 4 gives operating temperatures for these runs). Table 4 also gives laboratory-test data by Methods *(B)* and *(D)*. It is indicated that, because of the high temperature involved, test *(B)* emphasizes viscosity increase but does not reliably predict sludging stability. This is shown particularly by the results on oil *N*. In the engine, this oil reached the 20-mg. sludge-value at 42 hr. and, at 60 hr. contained 41.2 mg. of asphaltenes and had a viscosity of 172. By test *(B)* no significant amount of insoluble was formed in 12 hr., although the viscosity reached 390, and, by an extension of the test, the oil reached the sludging point at 13 hr., with a viscosity of about 500. Further, test *(B)* did not rate the four oils, as to viscosity increase, in the same order as found in the engine test. On the other hand it will be observed from Table 3 that test *(D)* did rate the oils in the proper order, with the exception of the minor variation between oils *N* and *C*, and also predicted the sludging behavior of oil *N*.

An inspection of Fig. 15 and, particularly of Table 2, shows that, in general, the laboratory sludging-time according to test *(C)* (305 deg. fahr. using oxygen) correlates with the corresponding engine sludging-time. There is, however,

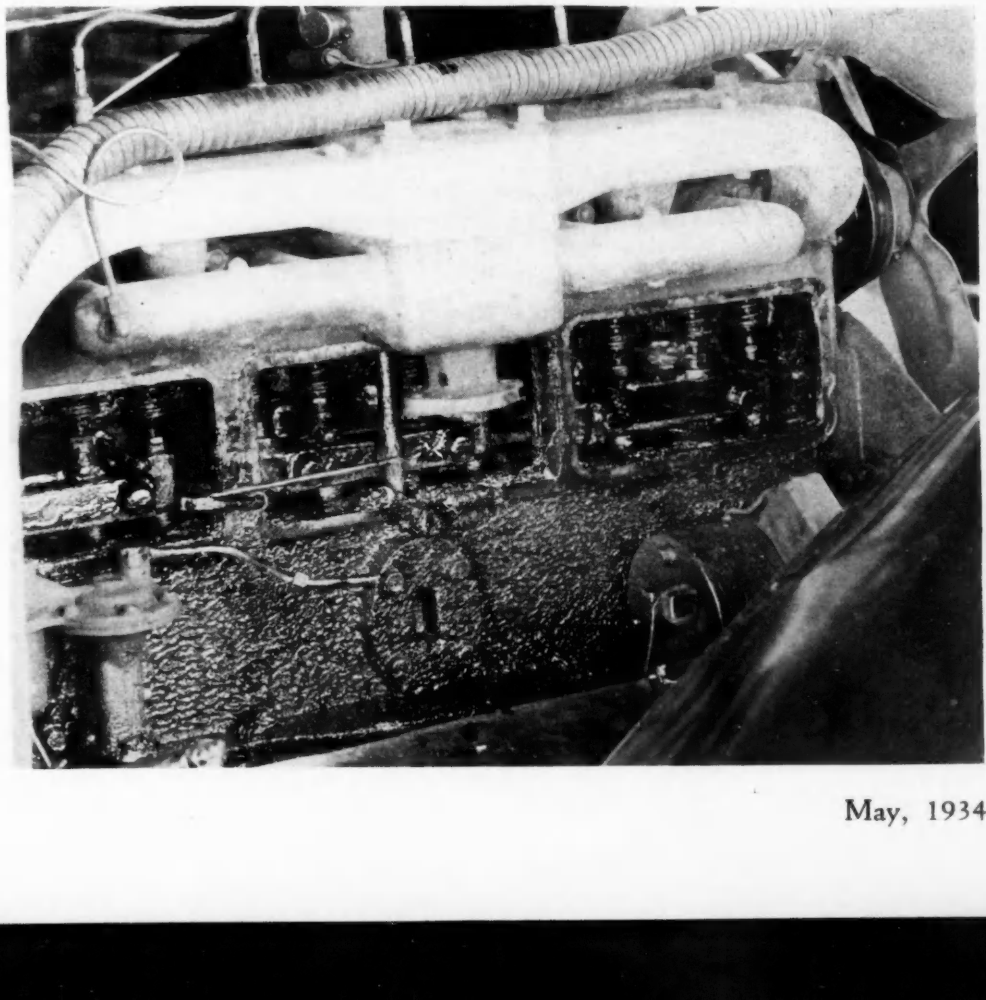


Fig. 6—Gelatinous Deposits in the Valve Compartments of a Water-Cooled Automobile Engine

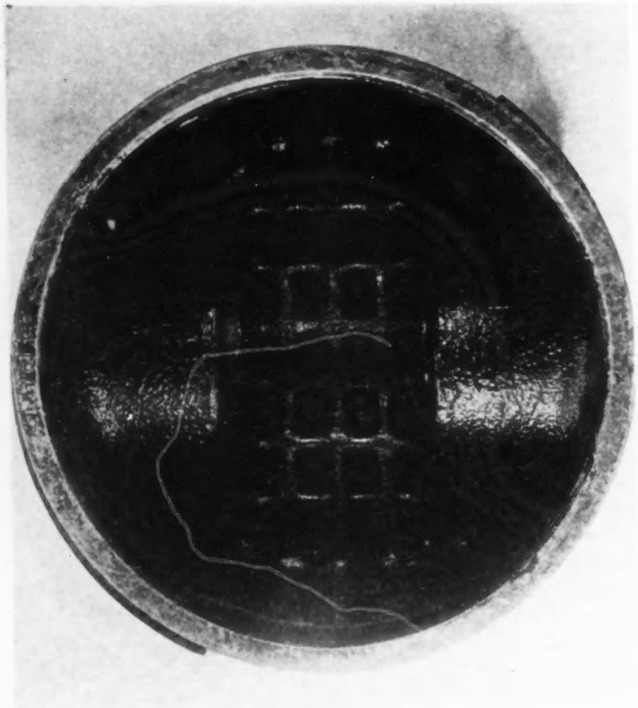


Fig. 7 *(above)*—Heavy Deposits on the Under Side of a Piston from an Airplane Engine

As opposed to the above, test *(D)*, using air at 341 deg. fahr., showed that the relative sludging-time of the two oils *A* and *F* differed markedly from that indicated by test *(C)*. The 341-deg. fahr. air-test, although rating many of the oils at approximately the same stability as test *(C)*, rates oil *F* in an entirely different class, and in line with the engine test. Extensive laboratory experiments have shown that oil *F* is not an anomaly. Due to the fact that the test using oxygen "punishes" an important class of oils unduly, it became obvious that the use of such a test could not afford a valid correlation with performance.

As shown by comparison of the sludging data in Table 2, the results of test *(D)* correlate well with engine performance. Many of the oils did not form significant amounts of sludge in the 50-hr. test, and a more elaborate comparison including viscosity change is given in a later section. Because test *(D)* correlated so reliably with available engine and service data, it has been adopted as a research tool by the authors' Laboratory and will be referred to as the "Indiana Oxidation Test." It is described in detail as follows:

Indiana Oxidation Test

As shown in Fig. 16, the oil under test is placed in a glass tube which is held in an oil bath regulated at constant temperature. The tubes are made of regular Pyrex tubing, 20 in.

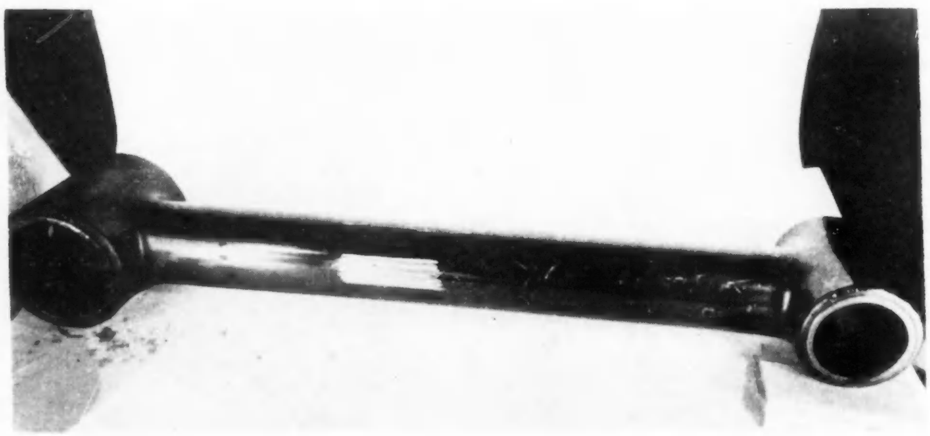


Fig. 8 *(right)*—"Lacquer-like" Deposits on a Connecting Rod from an Airplane Engine. The Lustre of This Particular Example Rivals That of the Finest Enamel

one marked discrepancy in the case of oil *F*. The two oils, *A* and *F* of engine tests Nos. 24 and 25, show the same laboratory stability, 31 to 32 hr., whereas the engine results are markedly different. The negligible asphaltene formation in run No. 24 indicates that the laboratory sludging-time of this oil should be over 150 hr. as was the case with tests Nos. 14, 16 and 19, which also showed negligible asphaltene formation in the engine.

long and 1¾ in. internal diameter. A flowmeter is provided to measure the stream of air which is delivered into the oxidation tube by means of a glass tube (3/16-in. internal-diameter), supported by a cork and reaching to within ¼ in. of the bottom. A "Bright Stock" of high flash-point and good oxidation stability is used for the bath oil.

The test is started with 300 cc. of oil, the level of which in the tube should be well below the bath level. The test oil

Table 3—Comparison of Oxidation Tests *B* and *D*

			Asphaltenes			Viscosity Increase			Volatility Loss, Per Cent
Oil	Run No.	Consumption	Engine 50 Hr.	Test *B* 12 Hr.	Test *D* 50 Hr.	Engine	Test *B*	Test *D*	Test *B*
N	108	0.154	30.1	Nil	20	53	271	30	6.1
V	137	0.164	Nil	Nil	Nil	24	215	17	8.9
C	136	0.178	Nil	Nil	Nil	56	193	29	8.6
Q	115	0.241	Nil	Nil	Nil	146[a]	145	41	9.6

[a] With a lower consumption, in line with the other three tests, it appears that the viscosity increase of oil *Q* would be 70-85 sec.

is kept at a temperature of 341 deg. fahr. (corrected), which requires a temperature of approximately 342 deg. fahr. in the bath. Air is passed through the oil at a rate of 10 liters per hr. (measured under laboratory conditions). Periodically, depending on the oxidation characteristics of the oil, 25 cc. of oil are removed, 10 grams of which are immediately weighed into a tared Erlenmeyer flask. This portion is diluted with 100 cc. A.S.T.M.-precipitation naphtha and allowed to stand 3 hr. before filtering through a prepared Gooch crucible. The crucible containing the insolubles is washed with approximately 100 cc. of naphtha, after which it is dried ½ hr. at 300 deg. fahr. and weighed. The amount of insoluble is expressed in milligrams per 10 grams of oil, and a sufficient number of samples are taken for test to determine accurately *(a)* sludging time, that is, the time required to form 10 mg. of naphtha insoluble per 10 grams of oil and *(b)*, the 100-mg. point, that is, the time to form 100 mg. of insolubles. These results are conveniently obtained by plotting insoluble against time of oxidation on a log.—log. chart.

The determination of viscosity increase, which becomes the more important criterion when sludging time is much over 50 hr., is made by taking a 100-cc. sample every 50 hr. This sample is promptly run for viscosity and put back in the oxidation tube. Oxidation tests are ordinarily continued either to the 100-mg. point or for 150 to 200 hr. for "sludgeless" oils.

Experience with this test by several laboratories has shown that the sludging time may be checked within about 5 per cent by different operators on different apparatus. Accurate temperature control, fairly exact control of air rate, change of the oil in the bath before marked thickening takes place and careful cleaning of the oxidation tubes, are essential. As it has been found that the sludging time varies inversely with the partial pressure of oxygen (within a limited range), correction must be made for tests at high altitudes. The tubes are cleaned by washing with naphtha, followed by soaking first with alcoholic KOH and then with chromic acid cleaning solution.

Correlation of the Indiana Oxidation Test with Performance, *(C)*

In the comprehensive study of the correlation of the Indiana test with performance, a second series of oils was run in an air-cooled engine. The technique of making these tests is described as follows:

Engine-Test Methods.—In making engine tests for sludge development, a definite procedure, based on a number of trial runs, was worked out. The method adopted, largely arbitrary, was adhered to as rigidly as possible in the making of all tests.

The engine used in these tests had the following specifications:

Number of Cylinders	6
Bore and Stroke	3½ in. x 4¾ in.
Piston Displacement	274 cu. in.
Compression Ratio	5.3
Horsepower, (N.A.C.C.)	29.4
Horsepower, (Rated)	100 at 3100 r.p.m.
Type	Air-cooled

Two engines were actually used. They had different cooling-fin arrangements but otherwise were identical. The data given in Table 2 were obtained on one engine while those in Table 4 were observed on the other. In each case the new engine was "run in" at light load and moderate speed, after which the load and speed were increased until the cylinder-head temperatures averaged about 450 deg. fahr. The engine was then dismantled and thoroughly cleaned, inspected and measured. After reassembly, several preliminary tests were made. During these tests temperatures in virtually every part of the engine were measured by means of thermocouples. Tests were run for 50 hr. with the engine delivering 24 b.hp. at 1500 r.p.m. Seventeen thermocouples were placed throughout the engine in each test. These, in conjunction with the cradle dynamometer, afforded a reliable and convenient method of checking and controlling test conditions.

The routine for each test was as follows: The engine was dismantled, all parts being removed except the crankshaft, camshaft and timing train. The individual parts were cleaned, and the carbon from the ring grooves and combustion chamber weighed. The engine was then reassembled, using new gaskets throughout, after which it was flushed with a mixture of 50 per cent benzol and 50 per cent alcohol by motoring with the dynamometer. Further flushing with oil from the supply to be tested was next in order. The crankcase was then filled with a weighed charge and the engine started.

After 15 min. of operation at light load, the engine was

Fig. 10—Heavy Deposits on a Piston from an Airplane Engine. The Material, While Smooth and Evenly Distributed, Is Somewhat Softer Than the "Lacquer-Like" Deposits Shown in Fig. 8 and Cannot Be Polished

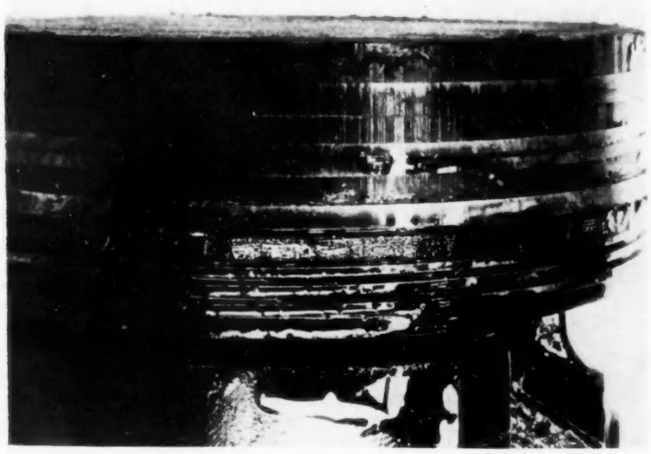


Fig. 9—An example of Ring Stocking Caused by Sludge. Note How the Heavy Deposits Have Been Dragged Over the Rings.

Table 4—Comparison of Engine and Indiana Oxidation Tests

Test No.	Oil	Engine Temperature, Deg. Fahr.: Sump	Engine Temperature, Deg. Fahr.: No. 3 Cylinder Head	Duration, Hr.	Oil Consumption, Lb. per Hr.	Rings Stuck	Final Insolubles (Mg. per 10 Grams): "Carbon"	Final Insolubles (Mg. per 10 Grams): Asphaltenes	Sludging Time, Hr.: Engine	Sludging Time, Hr.: Indiana Laboratory Test	Viscosity at 210 Deg. Fahr.: Original	Viscosity at 210 Deg. Fahr.: 50 Hr.	Viscosity at 210 Deg. Fahr.: 50 Hr. Laboratory
101	*A*	243	456	50	0.25	14	18.8	284	22	29	90	205	125
106	*L*	230	433	60	0.194	7	31.1	43.0	46	41	73	141	80
107	*M*	232	389	50	0.25	7	10.1	274	16	18	68	149	82
108	*N*	237	455	60	0.154	6	12.3	41.2	42	45	119	172	149
109	*A*	232	449	47	0.25	7	19.7	194	18	29	90	168	125
110	*C*	217	436	50	0.21	4	20.5	46.9	35	41	65	94	81
112	*A*	237	413	50	0.247	11	23.6	350	20	29	90	208	125
113	*P*	228	419	50	0.281	7	34.0	26.8	48	73	80	117	93
114	*C*	232	435	50	0.172	4	10.9	1.1	>50	>250	123	190	152
115	*Q*	247	436	48	0.241	3	3.8	0.5	>50	>350	124	270	165
116	*R*	228	462	50	0.206	8	11.7	4.5	>50	72	122	222	153
117	*S*	228	423	50	0.185	6	7.3	1.7	>50	>250	120	180	134
118	*T*	220	419	50	0.279	5	....	....	>50	>250	75	134	96
119	*U*	217	456	50	0.265	4	32.1	4.0	>50	104	87	110	102
133	*V*	244	465	50	0.274	7	19.7	3.0	>50	>250	120	172	137
136	*C*	236	449	50	0.178	5	(Total 3.1)		>50	>250	123	179	152
137	*V*	234	453	50	0.164	3	(Total 3.3)		>50	>250	120	144	137
144	*W*	236	452	50	0.164	8	9.9	3.8	>50	135	81	86	85

stopped for final valve adjustments. The test was then started, the engine being run continuously for 50 hr. except for a brief stop in the thirtieth hour to check the crankcase level, for which a calibrated gage-stick was used. No oil was added to the crankcase during the test. Samples were taken at intervals of 10 hr. and examined for acidity, total A.S.T.M.-precipitation of naphtha insoluble and the proportion of this which is chloroform soluble, and viscosity increase. At the end of a test, the remaining oil was drained and weighed. The consumption for the 50-hr. period was considered to be the difference between the original weight and that of the drainings, samples being regarded as consumed oil.

After the test, the engine was dismantled and the appearance of the parts noted. The condition of pistons and rings was recorded and carbon scraped from the various parts was collected and weighed. New parts were supplied to replace those broken or unfit for further service. Photographs were taken of various parts as a record of the appearance. Parts coated with the "lacquer-like" deposit previously described were buffed clean and bright.

It was found during the tests that, as the engine became worn from running and frequent cleaning, the oil consumption increased. When the consumption became too high to permit 50 hr. of operation without refilling, the engine was fitted with new rings and pistons. In one instance, regrinding of the cylinders was necessary to keep the oil consumption between the limits set. It would have been desirable, of course, to be able to make all tests at a fixed oil-consumption. This being impossible, particularly as the engine was a multi-cylindered one and also because of the rather wide range of oil viscosities encountered, it was found necessary to work as far as possible between certain limits which were taken roughly as 0.16 and 0.26 lb. per hr. (total consumption). Only a few tests were made outside this range; also, it will be noted that a few tests were made in which the 50-hr. period was not strictly adhered to. Not all of the engine tests that have been made are recorded in Tables 2 and 4. Many of those tests not recorded in this paper were simple duplicate or check tests. Others were those into which such variations had been introduced as to render the data not

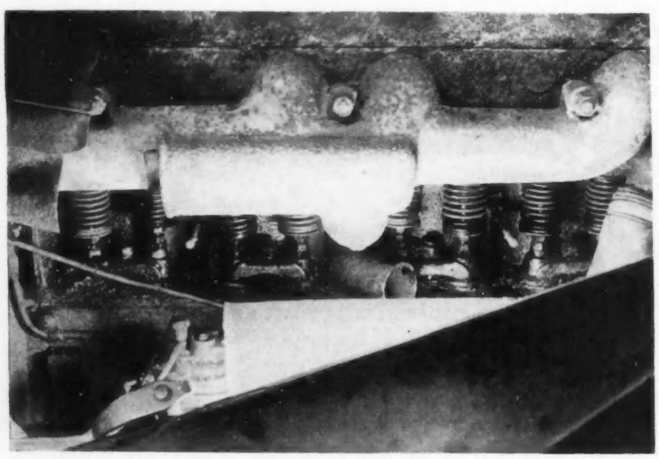


Fig. 11—Stiff "Putty-Like" Deposits in the Valve Chamber of an Elderly Automobile Engine

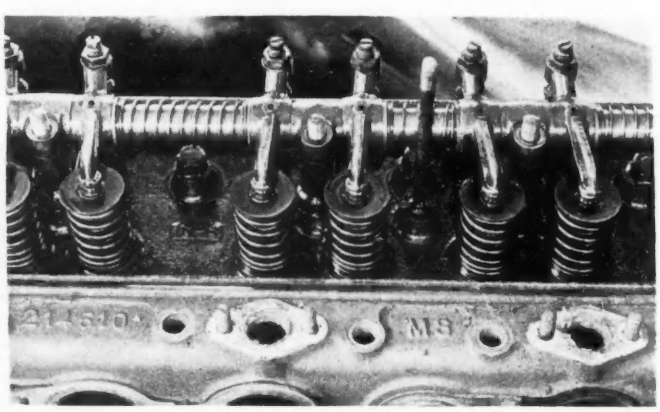


Fig. 12—Valve Gear of an Automobile Engine after 20,000 Miles of Road Operation on a Non-Sludging Oil. The Crankcase Was Not Drained During This Period, Nor Was the Filter Cartridge Changed. All Parts of the Engine Were Exceptionally Clean and Bright

pertinent to the present discussion.

By taking samples of the oil at 10-hr. periods, curves showing the rates of acidity formation, viscosity increase and asphaltene formation are obtained. The results of acidity determination were rather erratic and, in view of lack of evidence of any undesirable practical effects of acidity formation in itself, very little weight has been placed on these results, the usual observation being that the laboratory-test and engine-test results followed the same general trend. Viscosity increase becomes quite an important factor in some classes of oils, particularly those of the heavier grades which contain residual stocks. This class of oils is subject to oxidation at a rate even greater than light stocks; but no insoluble material is formed for a very long time, during which period the viscosity may increase to such an extent that the oil is unsatisfactory for further use.

Table 4 shows data on the engine tests and sludge and viscosity results on the corresponding oils when oxidized by the Indiana test. Inspection data on the oils used are given in Table 5. The temperatures of No. 3 cylinder head and of the oil sump are given as significant indexes of general engine temperatures. Oil consumption is indicated because it is an important consideration, the concentration of oxidized products being inversely proportional to the amount of the oil in the sump. The analysis and viscosity of the final (generally 50-hr.) oil sample are given, as well as the engine sludging-time as previously defined. It will be noted that the control oil *A* was run in tests Nos. 1, 9 and 12, to determine uniformity of engine conditions. The engine sludging-times of 22, 18 and 20 hr., respectively, give assurance that the results are reliable.

Fig. 17 shows a plot of the engine against the laboratory sludging-time. Widely varying types of oil—Coastal, Mid-Continent, Pennsylvania, as well as some whose method of refining makes them practically independent of crude source—are included, and there are no exceptions from a reasonably good correlation between oxidation-test results and performance. There are several oils which had not formed 20 mg. of asphaltenes after 50 hr. of engine service. The test is necessarily limited to about this length of time due to the oil consumption. Inasmuch as the operating conditions on this test are about as severe as can be imposed in practice, approaching the temperature limits at which the aluminum alloys involved can be safely used, it appears that the oils having engine sludging-times over 50 hr., corresponding to a laboratory sludging-time of more than 70 hr., will be substantially sludge-free in practice. With such oils, the addition of make-up oil becomes a governing factor in preventing insolubles from building up.

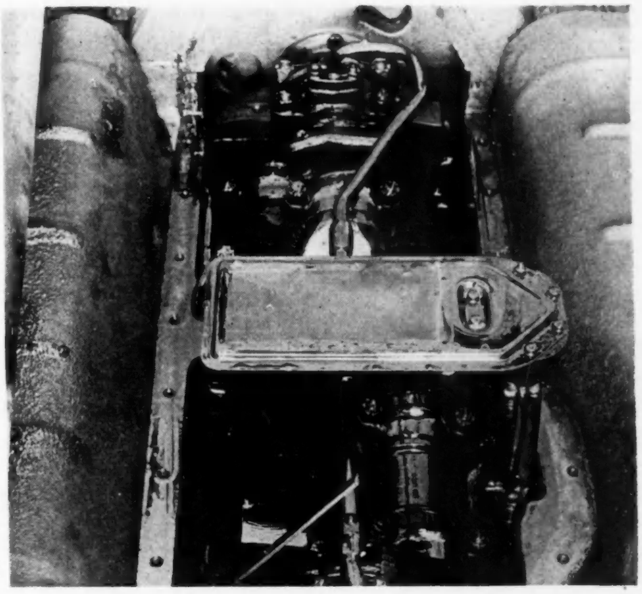


Fig. 13—Under Side of the Automobile Engine Shown in Fig. 12. Note the Exceptional Cleanliness of the Oil-Pump-Intake Screen. No Traces of Sludge Were Found in This Engine

A rather interesting comparison can be made with runs Nos. 13 and 16, two oils which have the same laboratory stability. The major difference of the oils is their viscosity, 80 and 122, which resulted in a lower consumption as is

Table 5—Inspection of Test Oils

Oil	Nature	Viscosity, Deg. Fahr. 210	Viscosity, Deg. Fahr. 100	Color A. S. T. M.	Flash Point, Deg. Fahr.	Gravity, A. P. I. Deg.	Pour Point, Deg. Fahr.
A	Commercial Motor	90	1565	3—½	470	22.0	0
B	Commercial Aviation	108	1283	2—2—½	510	31.1	—10
C	Commercial Aviation	123	1747	5—6	505	26.5	10
D	Experimental Highly Refined	114	1549	1—1—½	550	29.0	40
E	Commercial Aviation	158	....	7—½	....	....	..
F	Commercial Motor	85	860	8+	465	28.0	45
G	Commercial Motor	65	853	2—½	420	20.8	—10
H	Experimental	102	1681	4—4—½	500	23.6	25
J	Commercial Motor	92	....	3—½	....	....	..
K	Experimental	107	1624	4—½	480	....	..
L	Commercial Motor	73	617	......	....	....	25
M	Commercial Motor	68	894	3—	420	20.8	—10
N	Commercial Motor	117	....	4—½	490	23.0	20
O	Experimental	65	490	2—½—3	230	25.8	30
P	Experimental	80	845	3—3—½	410	26.4	40
Q	Commercial Aviation	124	1710	1—½—2	520	30.7	—15
R	Bright-Stock Blend	122	2275	7—8	510	23.9	5
S	Bright-Stock Blend	120	1680	5—6	530	27.8	0
T	Commercial Motor	75	....	2	450	30.5	—20
U	Experimental	87	1105	3—3—½	510	26.3	0
V	Experimental	120	1705	4	520	26.6	30
W	Commercial Motor	81	775	1—1—½	540	28.8	15

shown in Table 4. According to the indicated mechanism of oxidation in the engine it follows that the smaller residual amount of the lighter oil from run No. 13 should show a higher concentration of asphaltenes than does the more viscous oil. The 50-hr. asphaltene-values of 11.7 and 34 mg. thereby confirm this theory.

In the laboratory test, considerable attention has also been given to the rate of asphaltene formation after the sludging point is passed. This is obviously of considerable practical importance, although it is a difficult matter to get reliable engine data pertaining thereto, because of the effects on consumption produced by the insolubles previously formed. For this reason it has been feasible only to develop a correlation of initial sludging-times in laboratory test and engine test, which permits the assumption that laboratory sludging-rates are similarly valid.

Due to the fact that a fixed amount, representing a small proportion of the oil, undergoes oxidation in the engine at a given moment, it follows that the concentration of oxidation products obtained will vary with the amount of the oil in the sump. This is shown in Fig. 18, which gives results with a 34-lb. charge of oil compared with the normal 16-lb. charge. Similarly, the addition of make-up oil to keep a constant sump-level, which is ordinary service practice, has an effect upon asphaltene concentration which becomes quite significant after 25 hr. of operation. This is also shown in Fig. 18, where it is seen that, opposed to the steep rise in

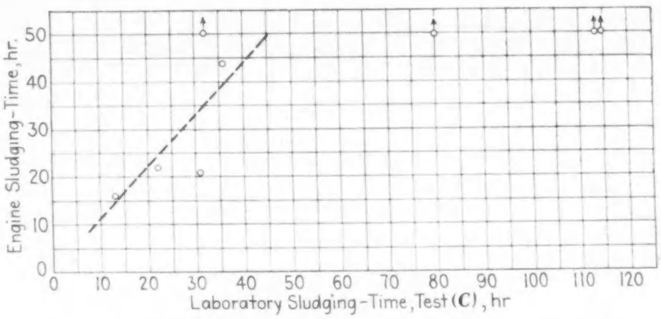


Fig. 15—The Extent of Correlation of the Laboratory Oxidation Test *(C)*, Using Oxygen at 305 Deg. Fahr., with the Engine Performance of Oils

The arrows indicate that the oil has not sludged at the completion of the engine test.

asphaltenes found near the end of the "dropping-level 16-lb. run," the "constant-level 16-lb. run" shows a flattening of the curve after long operation, due to the diluting effect of fresh oil. It can be shown mathematically that this curve approaches a maximum which is a function of the sludging time of the oil and the rate of addition. Thus, in practice, the consumption becomes an important factor governing the degree to which oil deterioration can go, and it is not to be anticipated that the very high asphaltene values found in some of these test runs would be obtained in actual operation.

From the foregoing discussion it is obvious that laboratory

Table 6—Properties of Service Tests Oils

Oil	Viscosity, Deg. Fahr. 210	Viscosity, Deg. Fahr. 100	A. S. T. M. Color	Flash Point, Deg. Fahr.	Gravity, A. P. I. Deg.	Pour Point, Deg. Fahr.	Sludging Time, Hr. (Indiana Oxidation Test)	Asphaltenes, 50-Hr. Engine Tests
A	90	1565	3—½	470	22.0	0	29	194–350
U	87	1105	3—3—½	510	26.3	0	104	4.0

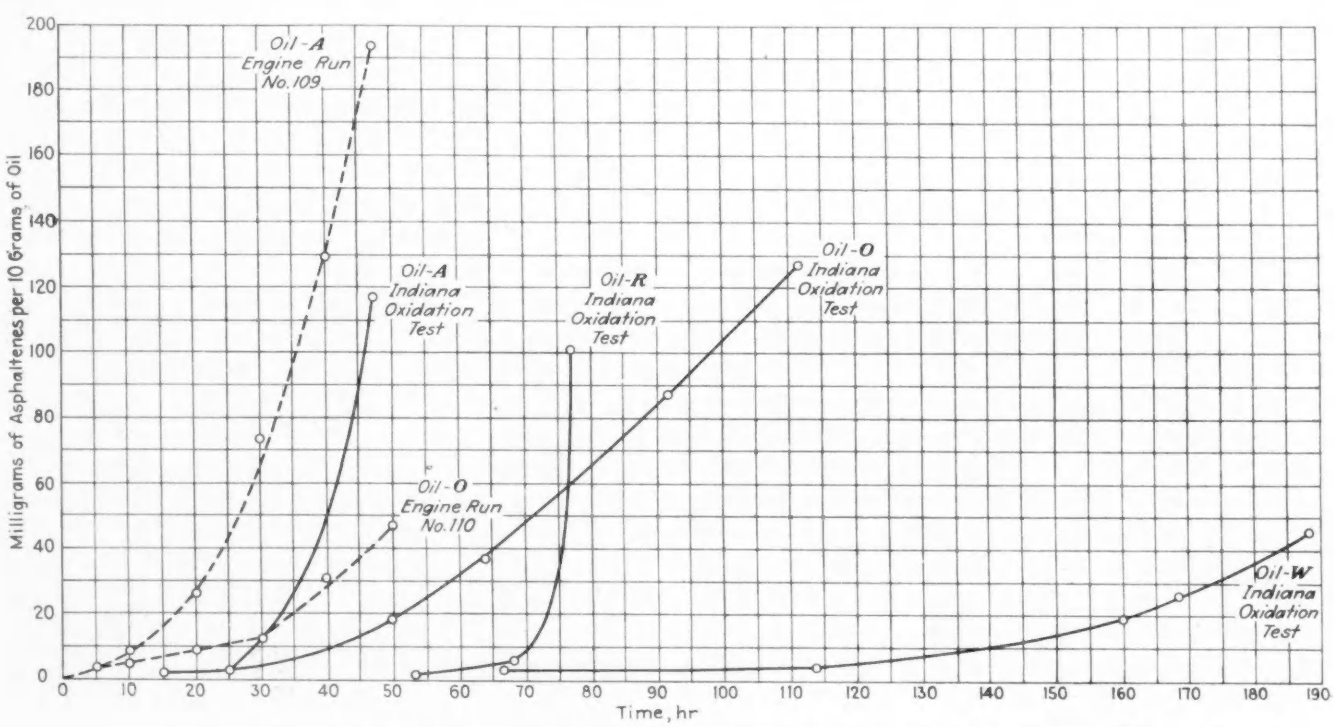


Fig. 14—Typical Sludging Curves of Oils Run in the Engine and in the Indiana Oxidation Test

Oil *W* showed only 3.8 mg. of asphaltenes at the completion of the 50-hr. engine-test.

sludging-times well above 70 hr. have little significance as far as comparative values are concerned and, hence, viscosity change becomes the primary consideration. In this respect, as shown in Figs. 19 and 20, the laboratory test is an adequate measure of performance. Fig. 20 shows viscosity after 50 hr. in the engine, the value being determined by extrapolation or interpolation in runs Nos. 7 and 9, compared to viscosity after 50 hr. in the laboratory test. In a correlation of this type it should be remembered that variations in oil consumption as well as in engine temperatures will influence the viscosity increase of the oil. The major discrepancies are run

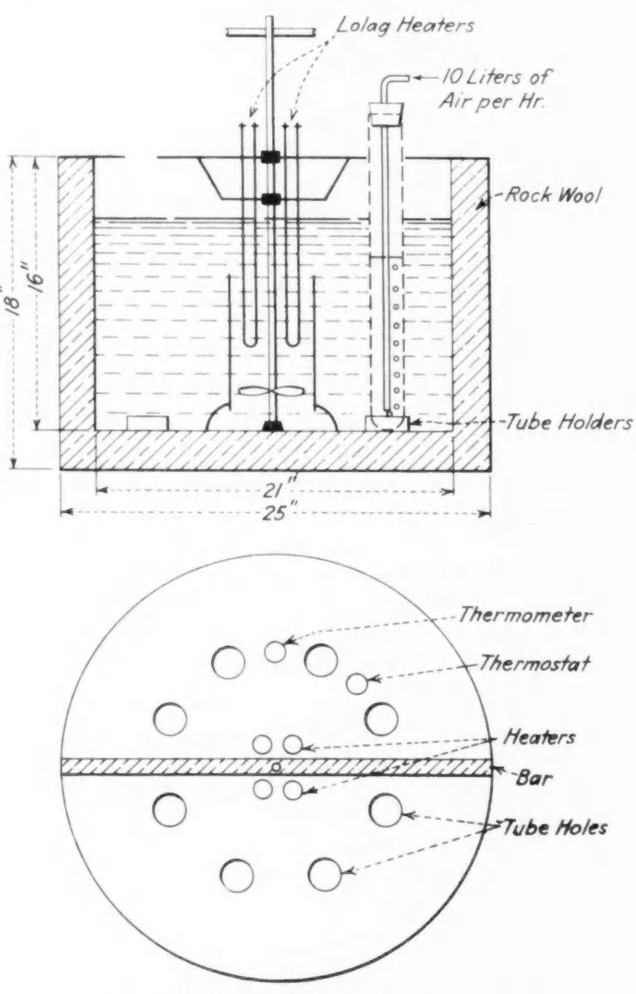


Fig. 16—Indiana-Oxidation-Test Apparatus

A cross-sectional view of the oil bath is shown at the top; the cover appears below.

No. 7, in which the engine oil-viscosity increase was unusually high, and run No. 8, in which it was extremely low. In general, considering the wide ranges of viscosity increase with the various oils in the engine and in the Indiana Oxidation Test, it can be said that the correlation is at least fair if not strictly quantitative. The increased oxidation of the oil in the engine toward the end of the run, when the oil-sump level is lower, explains this higher rate of viscosity increase. During the first 20 or 30 hr., the engine rate is only slightly higher than that of the laboratory.

Table 4 also shows the observation regarding ring sticking in each of the tests. It will be seen that, in general,

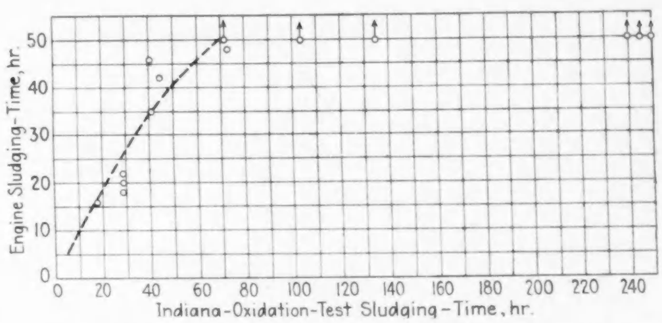


Fig. 17—Curve Showing the Correlation of the Engine and the Laboratory Sludging-Time with Various Types of Oils. Arrows Indicate That the Oil Has Not Sludged at the Completion of the 50-Hr. Engine-Test

tendency to ring sticking is a function of the sludging time found by the laboratory test. Those oils having stabilities less than 75 hr. averaged eight stuck rings, whereas the more stable oils averaged about five. As may be expected from the nature of the effect, the results as to actual number of rings stuck do not show the same quantitative correlation as in the case of asphaltene formation. One commercial, but generally unorthodox, oil has been found exceptional in that its ring-sticking behavior appears to be far worse than is indicated by laboratory sludging-time. Another specially processed oil is better than would be expected from the laboratory test. These latter facts indicate that tendency toward ring sticking may, under certain circumstances, be dependent on some other properties of the oil in addition to its sludging-time.

Service Observations

As the ultimate value of the laboratory work is determined by the results reflected in actual service, a series of observations was made on vehicles in fleet operation in which two oils which had shown laboratory engine-test behavior consistent with quite different oxidation-test values were employed. The tests were carried out without altering the operating schedules of the vehicles involved. The test procedure was quite simple, essentially being as follows:

The engines were thoroughly flushed with the oil to be tested and new filters were installed. After being placed in

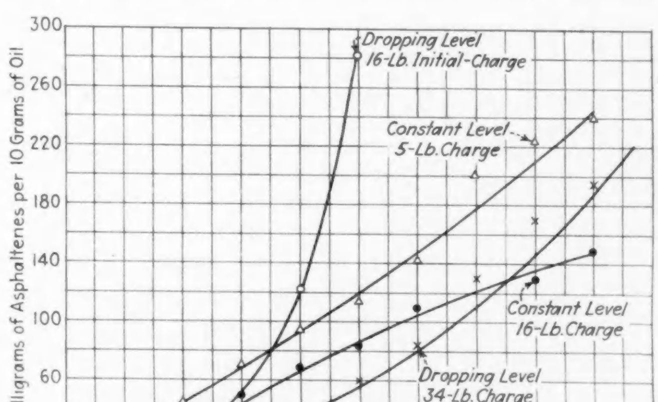


Fig. 18—Chart Showing That the Concentration of Oxidation Products Obtained Will Vary with the Amount of Oil in the Sump

The curves show the effect of varying oil-sump levels on the sludging characteristics of oil *A*.

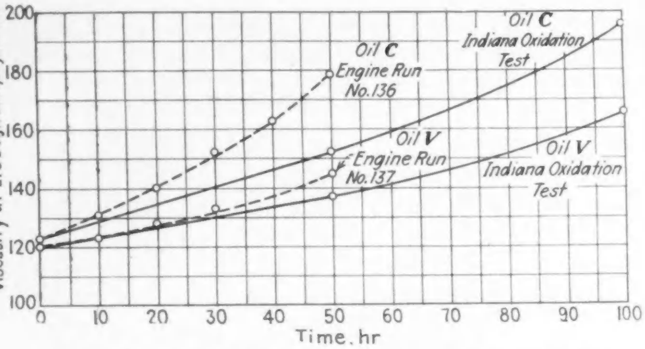


Fig. 19—Viscosity Development in the Engine and the Indiana Oxidation Test of Two Typical Oils

service, samples of oil were secured at each drain period for analysis, and deposits were taken from the oil filters and certain other selected areas. Observations on each oil were made for about six weeks, the procedure being repeated as often as possible for the period March 1 to Sept. 1, 1932. The crankcase samples were analyzed by the same procedure employed in the oxidation test. The deposit samples were made oil free by extraction with naphtha and the asphaltenes determined on the oil-free residue. Tables 6 and 7 give, respectively, the laboratory and the service data obtained on the two oils employed. Average values have been used in presenting the service data, as, of course, variations are much greater between individual observations than in the rigorously controlled laboratory tests.

In spite of the fact that the field-test data are not quite so clear cut as those obtained in the laboratory, it was consistently observed that the proportions of asphaltenes found in crankcase drainings and in engine and filter deposits were much less in the case of the stable oil *U* as compared with oil *A*, indicating that the former oil was oxidized to the lesser degree. These differences were somewhat masked, of course, by the fact that always some material from the previous observation period remained in the engine each time a new oil was introduced. With such an allowance it would appear that oil *U* is actually non-sludging, as the asphaltene concentrations, noted in Table 7, were quite small indeed. The tractor trucks undoubtedly represented the most severe type of service because of very high engine operating-temperatures, although the oil from motorcoaches in interurban service shows equal deterioration, probably due to greater mileage between drains.

The detailed data also indicated that oil *A* deteriorated more rapidly during the hot summer months than in the spring. A number of minor points were observed which, while in general qualitative in nature, tend to confirm the laboratory work. For example, where laboratory engine-tests showed that oil *A* resulted in filter deposits accumulating at about three times the rate for oil *U*, the average of a number of filter cartridges taken from the coaches used in city service showed total deposits of 50.6 and 20.2 grams on oils *A* and *U* respectively. The asphaltene concentrations found in these deposits

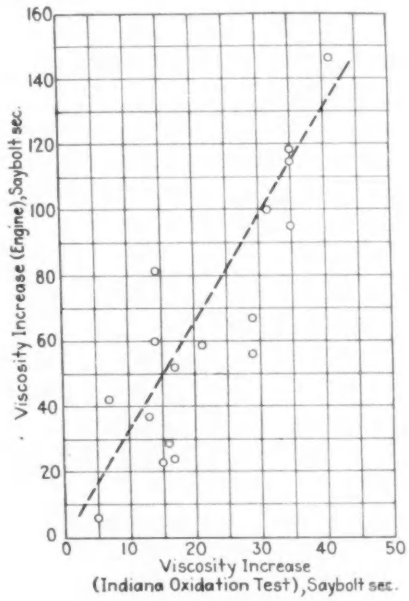


Fig. 20—Correlation of Viscosity Increase in the Engine and the Laboratory Indiana Oxidation Test

Viscosities are recorded as Saybolt-sec. increase at 210 deg. fahr. after 50 hr.

are given in Table 7. Further, even mechanics engaged in overhaul work reported visibly cleaner engines in the case of the stable oil.

In following the service tests, the attempt was made to determine whether the stable oil would lead to improved oil economy as predicted by laboratory data. The average values for consumption as given in Table 7 do show a trend in this direction. As to individual observations, however, it

(Continued on page 181)

Table 7—Service Tests

Equipment and Service	Oil	Consumption, Miles per Gal.	Miles per Drain	Used Motor Oil (Mg. per 10 Grams of Oil) "Carbon"	Used Motor Oil (Mg. per 10 Grams of Oil) Asphaltenes	Asphaltenes, Per Cent Crankcase Deposits	Asphaltenes, Per Cent Filters
Tractor Trucks—Freight	*U*	595	1500	30	3	8.0	11.0
4 Units, 156 Samples	*A*	325	1500	54	32	29.0	25.0
Motorcoaches—City Service	*U*	313	1500	20	6	....	16.0
28 Units, 478 Samples	*A*	296	1500	24	11	....	27.0
Motorcoaches—Interurban	*U*	608	2700	22	4	....	....
15 Units, 80 Samples	*A*	569	2700	24	39	....	....
Gasoline-Electric Rail-Car	*U*	136	1500	19	4	....	....
1 Unit, 32 Samples	*A*	88	1500	30	15	....	....

Various Methods of Analyzing Intake-Silencer Problems

By J. O. Almen and E. E. Wilson
General Motors Research Laboratories

INTRODUCTION of the closed body made the car silencing problem more difficult by adding "drumming" as another type of noise, it being a low-frequency noise periodic in character and associated with resonance phenomena. Studies on engine balance, engine mountings, intake and exhaust silencing, and related studies, were made by the General Motors Research Laboratories to determine the sources of the noises and provide means for their suppression.

Sources of periodic excitation of the body may be due (*a*) to vibration of the engine on its mountings and the transmission of such vibration to the body; (*b*) mechanical out-of-balance of any part of the drive train, the cooling fan and the tires; and (*c*) exhaust and intake-system resonance. After much previous research work, major attention was directed to the silencing of the intake and exhaust systems. An extensive laboratory investigation was conducted on "power roar"; but, after the noise sources in the intake system had been determined, their elimination seemed impracticable and it remained to develop a satisfactory type of silencer to be added to the conventional type of carburetor and manifold.

A description of this development is given, together with that of an exhaust silencer which also utilizes the principle of resonance silencing. In conclusion it is stated that the intake silencer has permitted a gain to be made in specific power per cubic inch of cylinder displacement as, by its use, the silencer has made possible a wider choice of valve timing without restriction by noise limitations.

NOISE suppression or silencing has taken a foremost place in car development due to the increasingly more exacting standards of excellence demanded. The present paper deals with one phase of this silencing problem; namely, the suppression of so-called "drumming" noises within closed bodies as caused by engine-intake noises.

The noise problem in automobiles did not become really acute until after more extensive use of closed bodies. In addition to accentuating, in most cases, the "mechanical" noises found in cars—such as rattles, knocks, gear noises and the like—introduction of the closed body has made the silencing problem more difficult by bringing into the picture another type of noise; that is, "drumming," which is a low-frequency noise periodic in character and associated with resonance phenomena. These periodic noises are especially pronounced in the closed body because of the formation of acoustic patterns within the body and resonance of the body panels.

As it became apparent that silencing of motor-car operation was to become a very important factor in engineering technique, the General Motors Research Laboratories early embarked on a series of studies to determine the source of these noises and provide means for their suppression. The investigation included studies on engine balance, engine mountings, intake and exhaust silencing, and other related studies. Sources of periodic excitation of body noises may be due to:

(1) Vibration of the engine on its mountings and the transmission of such vibrations to the body.

(2) Mechanical out-of-balance of any part of the drive train, the cooling fan and the tires.

(3) Exhaust and intake-system resonance.

Certain periodic noises were traced to engine vibration and lack of isolation of the engine from the frame. The harmonic crankshaft balancer was one of the General Motors Research contributions to better engine balance to aid in suppressing these noises and, likewise, the two-plane crankshaft as used on certain types of V-8 engines.

Very extensive work was done on engine mountings, which have been developed progressively as the public demand for

[This paper was presented at the 1934 Annual Meeting of the Society.]

better silenced automobiles has increased. The noises produced by vibration of the engine and its mountings were readily eliminated by reducing the frequency of the engine on its mountings below the driving range. This was accomplished by the use of progressively softer rubber mountings.

With the improvement in engine balance, better isolation of the engine from the chassis and body, better balance throughout the drive train, improved fan design and better body insulation, the lack of silencing in the exhaust and intake became more apparent and offensive. It began to be realized at this time that many of these periodic noises were often produced from different sources, even though, to the observer, they were of the same character. It became necessary then for the General Motors Research Laboratories to initiate a more elaborate program which would classify and separate out to a still greater degree the various noise sources which might be responsible for the body "periods" and to devise means for their suppression. Major attention has been directed to the silencing of the intake and exhaust systems.

Laboratory Investigation Conducted

As a result of preliminary study of intake noise, it was determined that the so-called "power roar," which was objectionable in certain types of automobiles, was a resonance condition in the intake system having a frequency on the order of 100-150 cycles per sec., and it was thought desirable to eliminate the noise at the source, if possible. An extensive laboratory investigation was conducted in an effort to determine the source of the disturbing intake sounds. It was noted that the intake noise was a maximum within relatively narrow engine-speed range at wide-open throttle and that the dominant note or pitch was unaltered whether the engine was running under its own power, whether it was motored on the dynamometer or being towed on the road. It was found that the dominant note persisted even though the carburetor and manifold were entirely removed. The source of the primary sound was, therefore, within the engine block, and apparently was resonance between firing frequency and cylinder frequency and was independent of the manifold.

Further investigation showed that the dominant note could be obtained with the engine stationary by blowing a jet of air over the intake port while the intake valves were open, thus indicating that the noise originated in the resonance chamber formed by the cylinder and valve which was excited by the flow of air through the valve at the time of initial valve opening. It should be noted that the frequency of this combination of valve opening and cylinder volume is not greatly changed with piston position because of the coincident change in valve opening. As is well known, a similar effect can be produced by blowing over the neck of a bottle, which will produce a tone dependent upon the volume of the bottle and the characteristics of the bottle neck.

The wide range of frequency in the intake system may be accounted for by the fact that the notes produced by the cylinder are amplified when approaching resonance with the frequency of the intake pulsations, and that the cylinder frequency in itself will vary over a narrow range depending on valve opening and the position of the piston in the cylinder. The function of the cylinder in producing intake noise likewise accounts for the effect of valve timing on intake-noise characteristics.

In addition to the particular tone produced by the cylinder, there was considerable influence of the intake manifold on the type of noise produced and its intensity. Likewise, some higher-pitched noises, above 200 cycles per sec., were amplified by the manifold to a considerable degree.

After the noise sources in the intake system had been determined, it became apparent that the elimination of the source of the noise was impracticable. It remained to develop a satisfactory type of silencer to be added to the conventional type of carburetor and manifold. At the outset, the ordinary practice of using a baffling arrangement was discarded because of the restrictive effects and because of the ordinarily poor performance of this type of silencer on relatively low-frequency noises. Of necessity, this silencer had to be especially designed for the suppression of this type of noise since, unlike the exhaust silencers, it was highly undersirable to introduce baffling of any description. This condition was necessary because of the much more serious effects of small restrictions in the intake on engine power than similar restrictions in the exhaust system. There remained, then, the possibility of using sound-absorbing material or of using a type of silencer which would produce the desired effect through the mechanics of resonance. It was soon found that the absorption type of silencer was limited in the range of frequencies which it would attenuate satisfactorily within the very limited space ordinarily available around the carburetor which could be allotted to a silencer.

Resonance Silencers Tried Out

A Quincke filter or tuning pipe was first tried which consisted of a dead-end tube, adjustable in length, placed at right angles to the intake passage and permitting unrestricted flow of air through the intake passage. It was soon found that this type of silencer was effective over much too narrow a band of frequencies. This limitation of the Quincke filter may be accounted for when the frequency characteristics of the intake system and the resonance characteristics of the Quincke filter are compared.

For example, assume a car in which an objectionable-noise period is perceptible from, say, 35 to 45 m.p.h. with maximum loudness occurring at about 40 m.p.h. This means that the frequency of the objectionable noise will cover a range of approximately 30 cycles per sec. corresponding to the change in the engine firing-frequency. The Quincke filter, on the other hand, has a comparatively sharp response and probably will not show appreciable silencing for a greater range than about 5 cycles per sec. at the outside.

Another limitation of the Quincke filter, in addition to the narrow silencing band, is the length required for attenuating the frequencies found in the intake system. With this type of silencer it is necessary for the closed-end pipe to be equal in length to one-quarter of the wave length of the sound to be suppressed. Since the frequencies of the noises to be attenuated are in the range of, say, between 100 to 150 cycles per sec., the corresponding tuning-pipe lengths will range from 36 to 24 in. Such lengths of pipe could not be fitted into the small space available without undesirable structural complications. Furthermore, if the range of frequencies is to be covered satisfactorily by the Quincke-type filter, a group of pipes would be necessary.

With the Quincke-type filter eliminated, other types of resonance silencers were then investigated with the result that the Helmholtz or resonance-chamber type of silencer seemed to offer the best possibilities. This type of resonator was chosen because its performance depended on volume rather than length, it could be made more compact and could be adapted to fit in the spaces allowable with much fewer constructional

difficulties, and it had a somewhat broader silencing band than the Quincke type; but still the range of frequencies covered by a single Helmholtz chamber was not sufficiently broad. Furthermore, it was noted that, in addition to the intake-noise period occurring in the speed range, say, from 35 to 45 m.p.h., an additional intake period was found to come in at another higher speed range with still other frequency characteristics. This required that at least two chambers would be necessary, and perhaps more, if the silencing range was to be sufficiently broad. To meet these requirements, the compound or series-type silencer, consisting of two chambers in series off the main sound-channel, was worked out. It was found that, for the same overall size of silencer, two ranges of frequencies could be covered with about the same silencing effectiveness as a single chamber covering only one of the frequency ranges.

Thus far we have considered only those noises issuing from the intake which were on the order of firing frequency. Frequencies which are some of the higher harmonics of the firing frequency likewise are present and must be silenced in order that a completely satisfactory job can be done. Some of these higher frequencies may be silenced by means of chambers or by means of sound absorbing material. The amount of silencing of the higher-pitched noises will depend on the balance between desired degree of attenuation on the one hand and cost and space limitations on the other. Ordinarily, the most practical silencer consists of a combination of chambers and sound-absorbing material, the latter being used primarily to silence high-frequency noises, such as hisses, originating in the carburetor. Other low-pitched noises issuing from the intake are most satisfactorily silenced by the chamber-type silencers.

Paralleling the development of the intake silencer, a satisfactory exhaust silencer, also utilizing the principle of resonance silencing, has been developed. One important result of these studies has been to show definitely that a design problem is presented in silencing either the exhaust or intake, which is as tangible as the design problem in, for example, a crankshaft. Furthermore, the results to be expected from a given design can be predicted by paper analysis.

Cooperative Design-Problem Studies

After the fundamental design-problems had been worked out for the resonance type of intake silencer, the commercial design-problems were worked out in cooperation with the AC Spark Plug Co. and with the Buick Motor Co. This cooperative study is being continued between the Research Laboratories and the manufacturing companies already mentioned, and most of the early difficulties are rapidly being overcome.

The chief difficulty still remaining is due to the lack of understanding on the part of the engine designer that the intake silencer is a part of a tuned system which must be as harmoniously tuned as a piano. For example, too many times a silencer is designed for a given engine but, when the silencer is tested, the expected results are not obtained simply because in the meantime a change has been made in, say, the valve timing, which to the engine designer seems quite unimportant but which has a very decided effect on the intake-noise characteristics. Therefore, the engineer in charge of engine development must realize the importance of seemingly minor changes, thereby saving much testing time. It is necessary in most cases that the engine should be set as to displacement, valve timing, carburetor and manifold, before a silencer is designed. Another common experience is to attempt to build a satisfactory silencer which is much too small for the volume of noise and range of frequencies to be suppressed. It must be realized that a silencer has a definite quantitative function to perform and must be dimensioned to its application as much as a propeller shaft or an axle shaft must be dimensioned to carry the loads imposed. The same may be said as well for the design of resonance exhaust-silencers.

It should be mentioned that though the silencer has the outward appearance of a tin can, it follows in its action very definite and well-known physical laws. These same silencers on intake or exhaust can exert very definite influences on engine performance, and often a poorly designed silencer will more than offset gains anticipated from an expensive change in engine construction. When these possibilities are completely appreciated by the designing engineers, the optimum in engine silencing, performance and economy may be fully realized. The intake silencer has permitted a gain to be made in specific power per cubic inch of cylinder displacement as, by its use, the silencer has made possible a wider choice of valve timing without restriction by noise limitations.

Causes and Effects of Sludge Formation in Motor Oils

(Continued from page 178)

must be said that they fluctuated quite widely indeed in common with general experience. Both consumption and asphaltene formation, as observed in service vehicles, might reasonably be expected to deviate quite widely from the actual laboratory values if for no other reason than that, whereas "make-up oil" is regularly added to the service engines, the laboratory work here recorded was all done without the addition of make-up oil and therefore with constantly dropping oil level.

Conclusions

To summarize briefly the work covered in this report, it is indicated that the formation of sludge in motor oils is due primarily to asphaltenes resulting from oxidation of the oil. Engine experiments indicated that the type of oxidation involved occurred at temperatures well above those normally existing in the oil reservoirs and that, in general, good agreement could be expected between representative engine performance and laboratory oxidation tests.

Of several laboratory tests investigated, the most promising consisted of a continuous oxidation test in air at 341 deg. fahr. The results of this test correlate well with carefully controlled heavy-duty-engine tests with respect to asphaltene formation and viscosity increase, and generally give sound indications of tendency to ring sticking. Extensive service tests with two oils differing widely in oxidation tests have served to prove the importance of oxidation stability, to confirm the conclusions of the laboratory work and to furnish further evidence of the effect of stability on oil consumption.

The authors wish to express their appreciation for the valuable contributory work of Kenneth Taylor, J. O. Eisinger, M. H. Arveson, M. L. Mack of the Research Department and H. R. Mathias of the Technical Division.

An Analysis of the Lysholm-Smith Hydraulic Torque-Converter

By Jarvis C. Marble
Patent Attorney, New York City

THIS converter is based on the turbine principle. The major parts consist of a pump wheel and a turbine wheel mounted to rotate in a common working chamber in a stationary casing in which the hydraulic or working fluid is circulated in a closed path of flow. The turbine wheel is usually provided with three rings or stages of blading working in conjunction with two rings of stationary guide blades carried by the casing.

Some of the statements made by Mr. Marble in connection with his description of the construction and performance of the converter are that one form, with reverse gear, is designed to transmit power continuously through the hydraulic mechanism; that reverse is effected mechanically, by spur gears, in one of the designs; that the chamber in which the working fluid circulates is maintained completely filled with fluid at all times during the converter's operation; that various fluids may be used as the working fluid; that the converter efficiency is relatively high over a comparatively wide range of speeds; and that the hydraulic efficiency reaches approximately 90 per cent.

After going further into construction detail and quoting performance data obtained from actual foreign service and service in buses of the International Railway Co., Buffalo, N. Y., Mr. Marble concludes his paper with performance statements made by the master mechanic of this company, Mr. Kunz, following block tests and operating tests of the converter.

THE Lysholm-Smith hydraulic torque-converter — referred to hereinafter simply as the converter—has been developed by Aktiebolaget Ljungstroms Angturbin, that is, the Ljungstrom Steam Turbine Co., of Stockholm, Sweden, to meet the long existing need in many fields of propulsion and power transmission for a true variable-speed transmission which is simple and rugged and which, from the standpoints of weight, cost of manufacture and maintenance, length of life and other practical factors, is comparable to the forms of transmission which have been employed up to the present.

The converter is based on the turbine principle. The major parts consist of a pump wheel and a turbine wheel mounted to rotate in a common working chamber in a stationary casing in which the hydraulic or working fluid is circulated in a closed path of flow. The turbine wheel is usually provided with three rings or stages of blading working in conjunction with two rings of stationary guide blades carried by the casing.

Fig. 1 shows in central longitudinal section one form of the converter, with reverse gear, designed to transmit power continuously through the hydraulic mechanism. The pump wheel, which may be driven directly from the engine, has a single ring of pump or impeller blades *a* which deliver the working fluid radially and outwardly to the first ring of blades *b* carried by the turbine wheel. From the first ring of turbine blades the fluid passes through a first row of stationary guide blades *c*, which serve to reverse the rotational component of flow and deliver the fluid to the second ring of turbine blades *d*. From the second ring of turbine blades the fluid passes through a second ring of stationary guide blades *e*, and finally passes through the last ring of turbine blades *f* from which it is returned to the suction side of the pump blades.

Fig. 2 shows the major component parts of the hydraulic mechanism, disassembled. The pump wheel and front-casing half appear at the left, the turbine wheel at the center and the rear-casing half, with its two rows of guide blades, at the right. Reverse is effected mechanically and, in the design illustrated in Fig. 1, a conventional spur-gear reverse is employed.

The chamber in which the working fluid circulates is maintained completely filled with fluid at all times during

[This paper was presented at the 1934 Annual Meeting of the Society.]

operation of the converter, and a small expansion-tank is connected to this chamber for holding excess and reserve fluid. To maintain the fluid in the working chamber under pressure, a small jet-pump or ejector arrangement is employed which utilizes the pressure of fluid taken from the working chamber on the discharge side of the pump. This jet pump keeps the working chamber filled with fluid taken from the expansion tank and keeps the fluid on the suction side of the pump under positive pressure. By this arrangement cavitation on the suction side of the pump, which would adversely affect efficiency, is prevented. The maintenance of the working chamber completely filled with working fluid at all times when the converter is in operation contributes to the absence of wear in the blading.

Various fluids may be used as the working fluid for the converter; but, up to the present, the fluid usually employed has been kerosene to which has been added a small percentage of lubricating oil.

Leakage of fluid from the working chamber is slight, since the seals for retaining it are of small diameter at the center of the converter and subject only to relatively low pressure. Metal-to-metal seals are employed and, in some designs, the faces of the seals are deliberately scored to provide lubrication by the working fluid. The small amount of fluid passing the seals drains to a sump from which it is returned to the expansion tank by an ejector, which is also operated by fluid pressure derived from the working chamber of the converter.

Due to the relatively high efficiency of the converter, the amount of heat generated is comparatively small and this heat is dissipated by means of a small radiator through which circulation is obtained due to the difference in pressure between the discharge and suction sides of the pump. By thus utilizing the available fluid pressure, all auxiliary functions are taken care of without the necessity for additional moving parts.

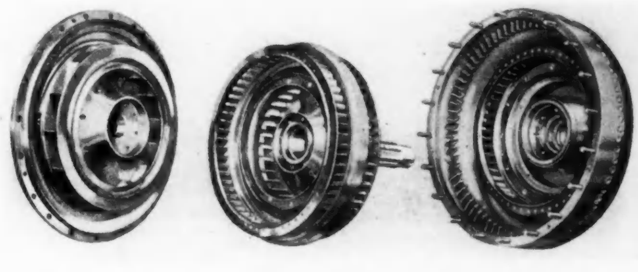


Fig. 2—Major Component Parts of the Hydraulic Mechanism

Fig. 3 shows characteristic torque and efficiency curves with relation to the secondary or driven-shaft speed for a converter operated by a gasoline engine at full throttle. Maximum torque multiplication for this type of the converter is approximately 5:1. At idling engine speeds the torque transmitted to the driven shaft is too small to overcome the inertia of a vehicle; a clutch is therefore not required.

The efficiency of the converter is relatively high over a comparatively wide range of speeds, and the hydraulic efficiency reaches approximately 90 per cent. In the first converter built, the hydraulic efficiency obtained by test reached 88.7 per cent. The high efficiency and the comparatively flat character of the efficiency curve is attributable to a number of factors, among the most important of which are the number and arrangement of the stages of pump and turbine blading, the continuous maintenance of a sufficient positive

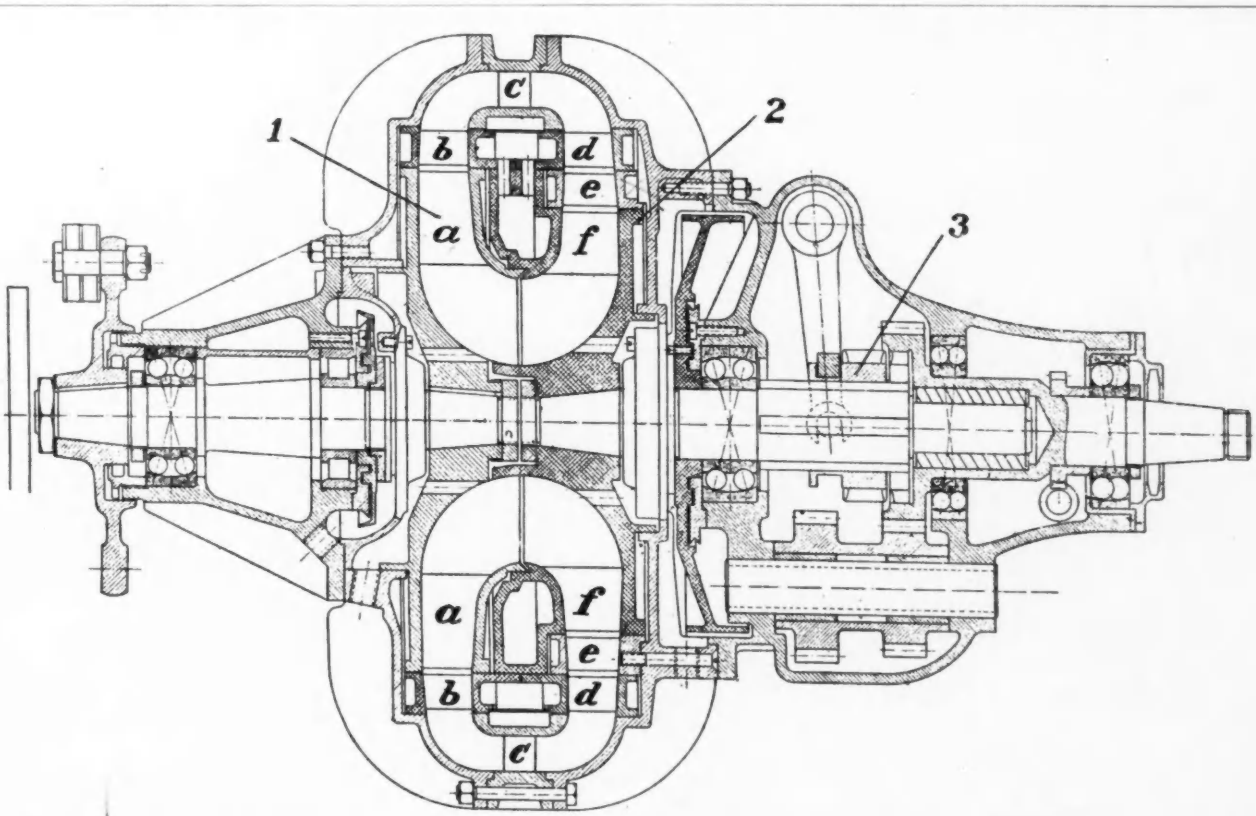


Fig. 1—Lysholm-Smith Converter with Reverse Gear

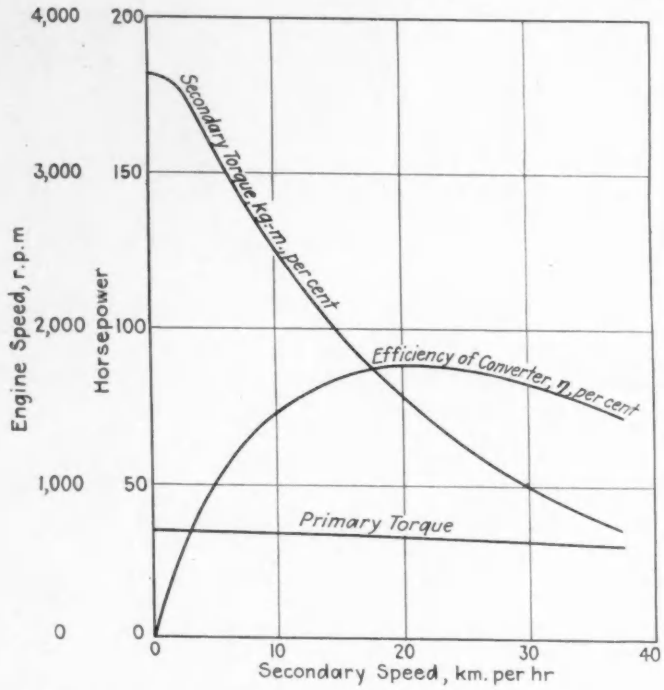


Fig. 3—Torque Curves of a Lysholm-Smith Converter

pressure of fluid in the working chamber of the converter and the special profile of the blades.

Because of the variable speed-ratio, it is impossible to obtain equally favorable flow conditions in the converter at all speeds. With blades of normal profile, even a slight variation of the inlet angle of the fluid with respect to the blades will materially affect the efficiency because of the introduction of substantial eddy losses. To avoid this, blades having very pronouncedly blunt rounded inlet edges are used, and the adverse effect on the efficiency of the converter due to changes in the inlet angle of the working fluid is comparatively small with these blades.

Provision for Direct Drive

In numerous fields of application it is desirable to provide for direct drive between the driving and driven shafts, and Fig. 4 illustrates a type of converter designed to provide positive direct drive in addition to the variable-speed hydraulic-drive. The hydraulic end of this converter is the same as that shown in Fig. 1, except that the pump shaft *B* and the turbine shaft *E* are hollow. A shaft *L* for direct-drive power-transmission passes through these hollow shafts. Power is transmitted through the medium of friction clutches alternatively to the pump shaft *B* of the hydraulic mechanism or to the direct-drive shaft *L*.

In the design illustrated, the friction-clutch mechanism consists of two driven discs *A* and *K*, the former connected to the pump drive-shaft and the latter to the direct drive-shaft. The driven discs are housed in a box-type housing which may form the engine flywheel, and are alternatively engaged by a common pressure plate *M* located between the two driven discs. Shifting of the pressure plate to one or the other of its positions of engagement is effected by a simple toggle mechanism, the pressure plate being held in engagement by a spring or springs acting on the toggles.

Control of the position of the pressure plate may be effected manually, by vacuum servo-motor control, or by any other type of control means dictated by the needs of the specific installation. This clutch control also provides the possibility for a neutral position, if such is desired, in which the pressure plate is out of engagement with both of the driven discs.

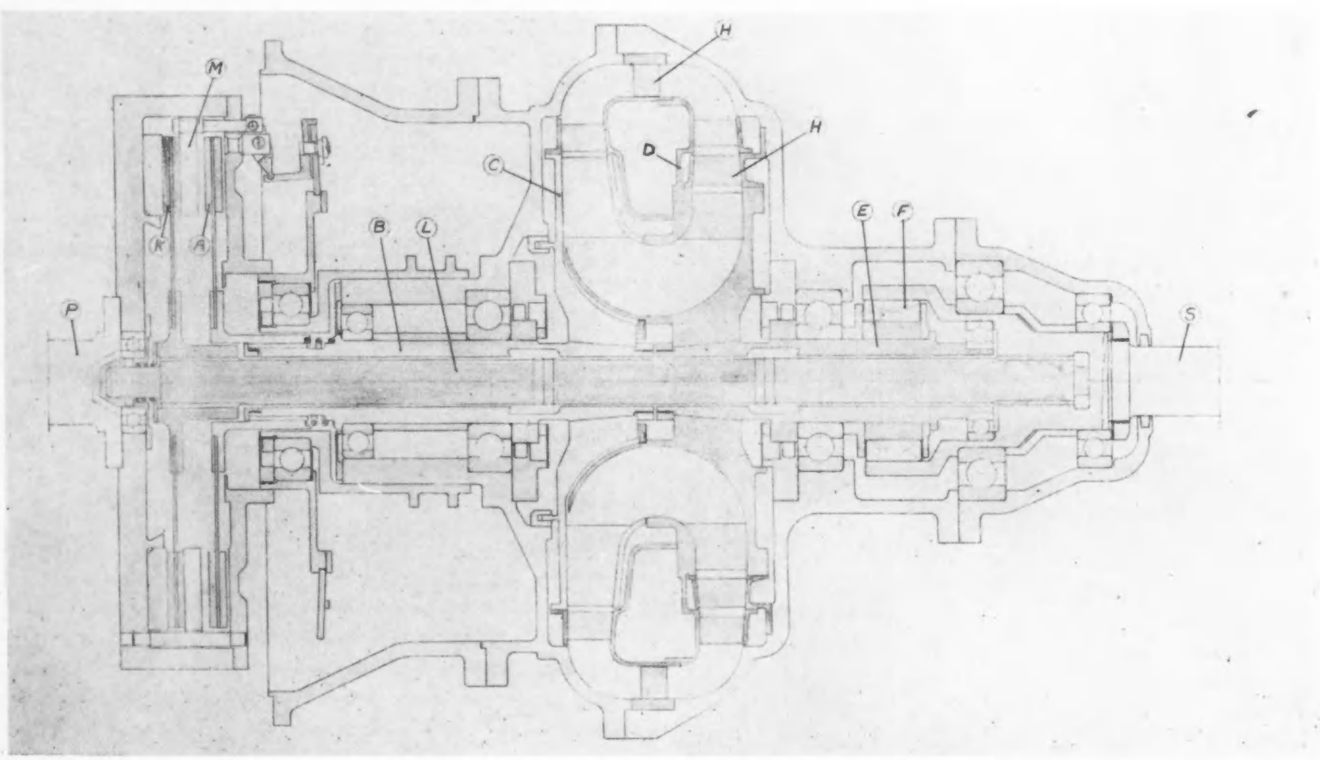


Fig. 4—Lysholm-Smith Converter with Mechanical Direct-Drive

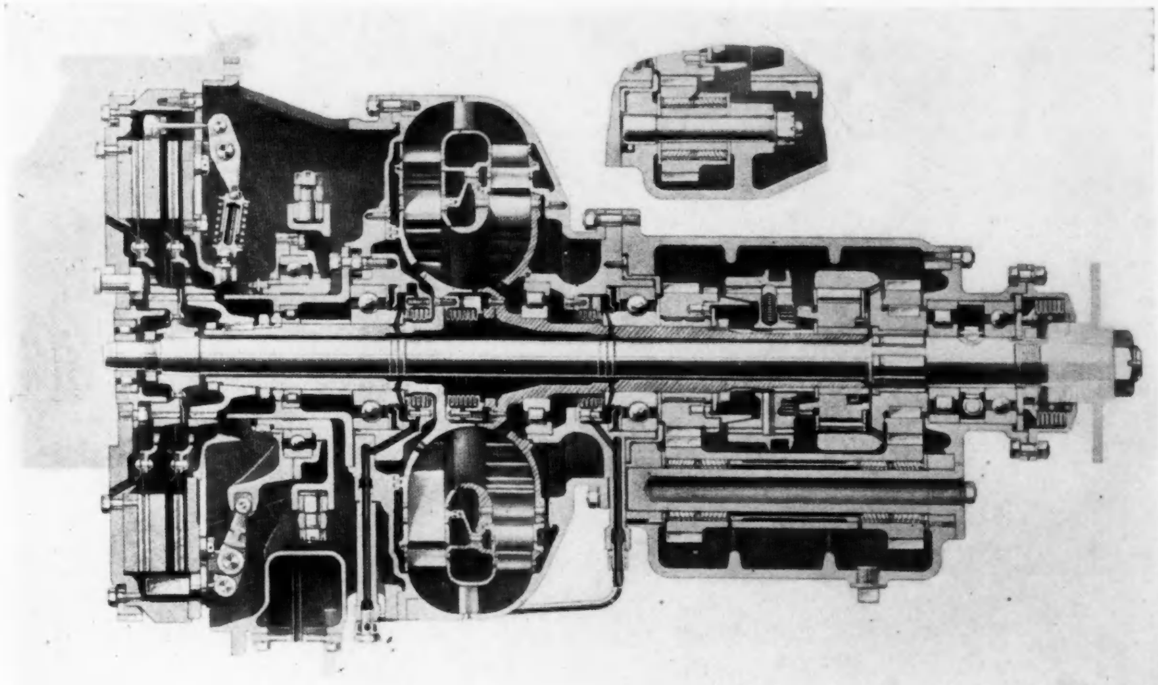


Fig. 6—Leyland Motors, Ltd., Bus-Type Lysholm-Smith Converter Having Direct Drive and Spur-Gear Reverse with a Synchronizer

The clutch plates used in this arrangement are not subject to the wear encountered with clutches used in conjunction with gear transmissions, since, with the converter, the load is not picked up by the clutch.

In the type having the direct-drive feature, drive to the propeller or other driven shaft *S* is transmitted from the turbine shaft *E* of the hydraulic mechanism through an overrunning or free-wheel clutch *F* of the usual roller type. This is done to permit the turbine member as well as the pump member of the hydraulic mechanism to come to rest when direct drive is being employed, thus avoiding any hydraulic drag when in direct drive.

Fig. 5 illustrates torque and efficiency characteristics for an 85-hp. bus-installation in which both hydraulic and direct drives are employed. With the direct-drive arrangement any suitable type of reverse gear may be used, and Fig. 6 shows a converter which is now being manufactured on a commercial basis in England, complete with a reverse gear of the spur type.

The direct drive provides positive mechanical connection between the engine and the driven shaft, which, for many practical reasons, is highly desirable. In addition, the converter can be constructed so that a very powerful engine-actuated hydraulic-brake is available at all times, since the nature of the hydraulic mechanism permits the converter to be thrown into reverse without shock with the engine running and regardless of the direction of rotation of the driven shaft. To obtain the benefits of this hydraulic braking, a planetary reverse gear with friction locking is employed, which permits engagement of the reverse gear at any time. Two types of reverse gear are at present in use in different applications and, with the spur-gear reverse shown in Fig. 6, a synchronizer is employed to facilitate engagement of the reverse gear with the shafts in motion.

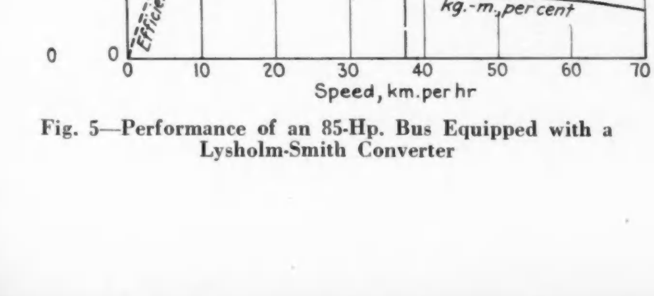


Fig. 5—Performance of an 85-Hp. Bus Equipped with a Lysholm-Smith Converter

Broad Field Open to the Converter

The field of application for the converter is very broad, ranging from passenger-car and light-commercial-vehicle use to heavy installations for marine, railcar and locomotive drives. In some fields of application, the type of converter shown in Fig. 7 can be employed to advantage.

In the type of converter shown in Fig. 7, the turbine and stationary guide blades are like those in the types previously discussed, but the pump blades are pivoted instead of being fixed. The design illustrated provides a relatively simple and rugged means for turning the pump blades by shifting the position of an actuating shaft *B* rotationally with respect to the hollow pump-shaft *A*. The shift is accomplished by axial movement of the collar *P* which is splined on shaft *A* and provided with diagonal slots into which project the ends of the transverse pin *N*. The pin *N* is fixed in shaft *B* and passes through peripherally extending slots in shaft *A*. The pump blades are attached to discs *H* set in recesses in the face of the pump hub *E* and the discs and blades pivot about the pivot pins *K*. Shaft *B* carries a hub plate *F* having teeth meshing with teeth on the discs *H*, thus causing the discs and pump blades to turn about the pivot pins *K* when plate *F* is turned relative to the hub *E*.

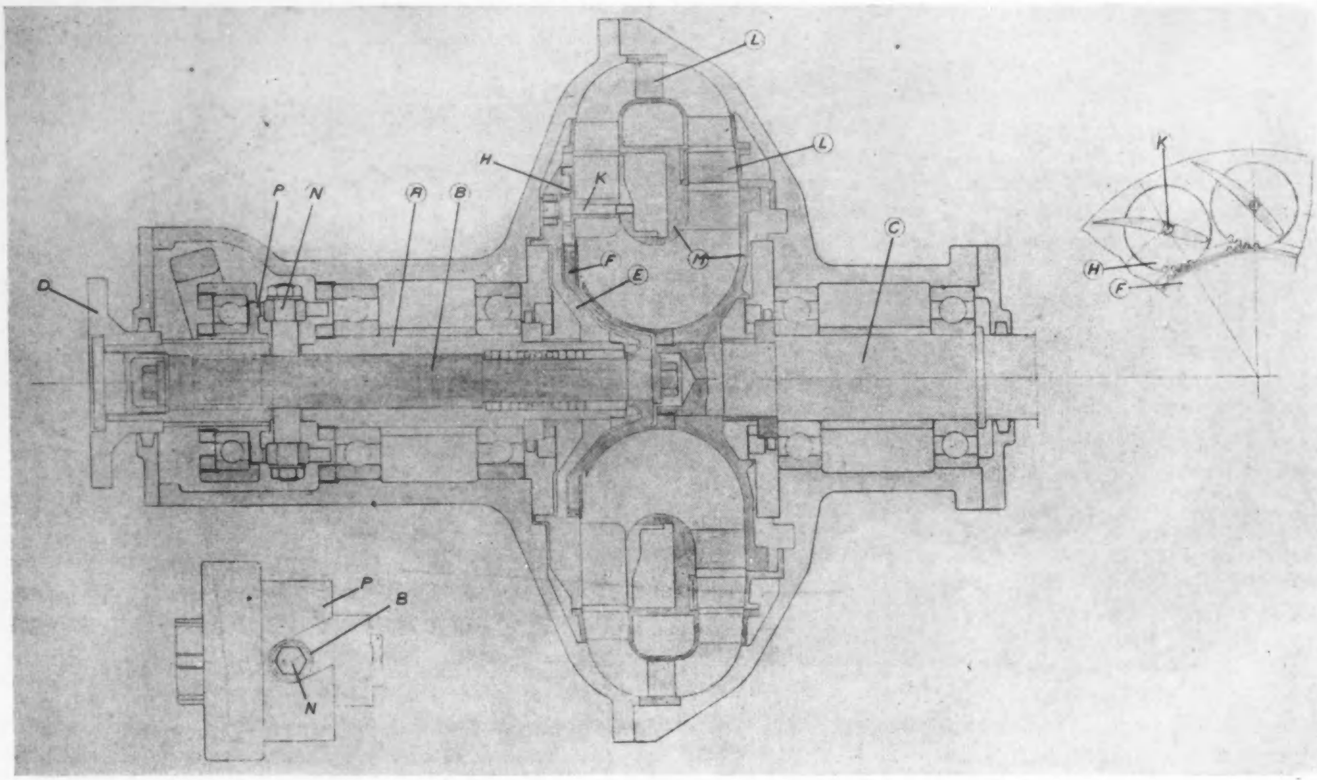


Fig. 7—A Lysholm-Smith Converter with Adjustable Pump-Blades

With movable pump blades a higher maximum torque-multiplication can be obtained than with fixed pump-blades, and the speed range over which high efficiency of operation is obtained may be made even greater than with the type of converter in which fixed pump-blades are employed. By closing the pump blades, circulation of the working fluid is stopped and a neutral position is obtained without using a separate clutch in which no load is imposed on the engine.

Converter Size and Weight

The size and weight of the converter required to transmit a given amount of power depend upon the speed at which the pump and turbine elements operate. A converter capable of transmitting around 125 to 130 hp. at 2000 r.p.m. will have an overall diameter of about 18 in. and the overall length of such a converter, having the direct drive feature and reverse gear of the kind illustrated in Fig. 6, is about 32 in. With planetary reverse gear the overall length of such a converter may be materially less. A converter for transmitting approximately 70 hp. at 3600 r.p.m. will have an overall diameter of the hydraulic mechanism of about 13 in., and the overall length of the unit type for direct connection to an engine and including direct drive and reverse gear is about 18 in. from the rear face-plate of the engine crankshaft to the face of the companion flange for the universal joint at the rear of the converter.

The weight of the converter is comparable to the weight of the clutch and gear mechanism which it replaces. The nature of the converter makes it readily adapted to mass-production methods. The clearances may be relatively large and tolerances are well within the limits of usual modern automotive standards. The larger the size of the converter, the easier it is to obtain high hydraulic efficiency. For a converter designed to transmit from 150 to 250 hp., it is possible to increase the efficiency to a value over 90 per cent.

All of the major parts of the pump and turbine rotors are circular and are easily turned out. The profile of the blading is such that it can be drawn or extruded to practically finished profile, thus eliminating expensive milling operations in producing the blading in its finished form.

Experience has proved wear in the blading to be a negligible factor, and converters which have been examined after many thousands of miles of service have shown no perceptible deterioration in the hydraulic parts of the apparatus. The life of these parts, assuming that the converter receives reasonable attention, appears to be of indefinite length.

The characteristics of the torque and efficiency curves can be altered very materially to fit the converter to the requirements of different specific installations. For railcars, for example, in which engines of relatively large power are employed and in which it may be desirable to dispense with the direct drive, the hydraulic mechanism can be designed so that the peak of the efficiency curve is reached at the average normal operating-speed, while at the same time retaining torque-multiplication characteristics satisfactory for this type of service. Both the fixed and movable pump-blade types of converter are applicable in different kinds of railcar service.

The converter in its present state represents a number of years of intensive development work which has brought it to commercial practicability. It is not possible here to review all of the practical applications that have been made, but some of the experience indicative of the present state of development is summarized as follows:

As early as 1928 a converter had been built and successfully operated in a 50-hp. bus-chassis of Swedish manufacture. In Stockholm, a converter has been operated for some time

in a bus of the Stockholm Tramway Co., and the average gasoline mileage obtained over 11,060 miles of operation was 7.35 miles per gal. Two other buses of the same type but equipped with gear transmissions averaged respectively 7.08 miles per gal. over 32,300 miles of operation, and 6.74 miles per gal. over 18,000 miles of operation. The bus equipped with the converter is being operated under the most severe service conditions of the three.

The converter was exhibited at the Olympia Show in London in the fall of 1931 by Leyland Motors, Ltd., licensees for the British Empire in the bus, truck and railcar fields. Since that time the converter has been exhaustively tested by the Leyland company, which now has actual operating experience with the converter amounting to several hundred thousand bus miles. During 1933, Leyland prepared to manufacture the converter on a production basis, and has adopted it as a standard equipment in its buses and railcars. The converter, as built by Leyland, is carried as a unit with the engine and a typical Leyland installation for buses is shown in Fig. 8, in which the size of the converter relative to that of the engine is indicated, this being such that the converter can readily be installed in the ordinary chassis in the space usually occupied by clutch and gear transmission. The converter shown in section in Fig. 6 is the same as that illustrated in Fig. 8.

Fig. 9 shows the relative rates of acceleration obtained by Leyland with its Titan bus equipped with the converter, and equipped with gear transmission. The Titan bus is a large double-deck omnibus type, and the acceleration curves represent the acceleration obtained with a fully laden converter-equipped bus and a lightly laden bus equipped with gear transmission. According to Leyland, the acceleration rate shown for the gear-equipped bus can hardly be improved upon without undue discomfort to passengers.

Early in 1933, a 200-hp. converter-equipped railcar supplied by Leyland was placed in service in Ireland by the London, Midland & Scottish Railway. This railcar is provided with two driving units of the kind illustrated in Fig. 10. The type of converter employed provides direct drive, the hydraulic mechanism and clutches being carried as a unit with the engine and the reverse gear being incorporated in the gearbox at the driving axle. This equipment has proved to be very satisfactory in service and, recently, Leyland has received orders for additional units of this kind from the aforesaid railway.

Converter Use in the United States

Up to the present, the greatest amount of practical operating experience with the converter in the United States has been obtained with a converter installed and operated in buses of the International Railway Co., in Buffalo, N. Y. This converter, of Swedish manufacture, is of the same type and has approximately the same power-transmitting capacity as the bus converter now being offered by the Leyland company in England. It was first installed at Buffalo early in 1933 in a relatively old bus having solid-tire equipment. This bus had been equipped with a gear transmission and, for some months after the installation of the converter, the bus was operated only in rush-hour service.

The primary purpose of this original installation was to determine not only comparative performance but also the ability of the converter to stand up under the severe vibration and shock to which it was subjected with this solid-tire equipment. After some months of service, which was quite satisfactory and in which the converter was operated for a total of 4789 miles, the converter was removed and installed in the bus in which it is now running in regular service. This bus is a Yellow Coach Model Y-Z gasoline-electric vehicle, powered by a Model Y-Z Knight sleeve-valve six-cylinder engine of 4¼ x 5½-in. bore and stroke, developing approximately 90 hp. at 1800 r.p.m. In making this installation, the only changes made in the bus were the removal of

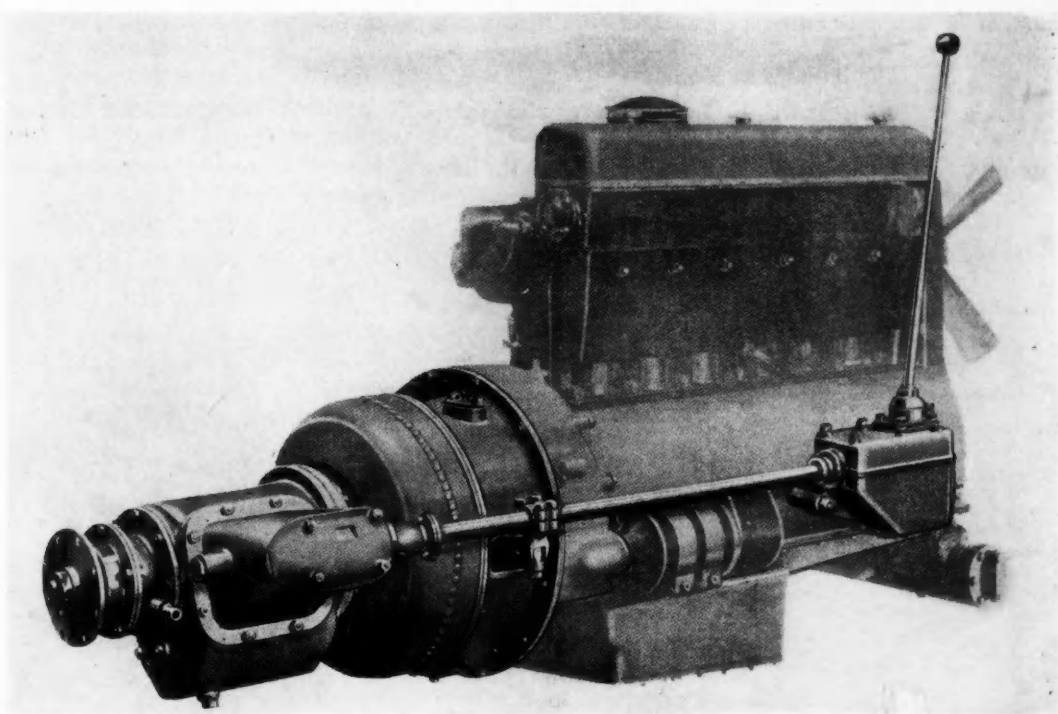


Fig. 8—A Lysholm-Smith Converter as Built by Leyland Motors, Ltd., for Bus Application

the electric generator and motors and the substitution of a single-drive rear-axle having a 7:1 reduction ratio in place of the double drive-axle used with the electric equipment. The light weight of the bus with electric drive was 16,550 lb. After conversion to hydraulic drive it was 15,400 lb., making a net saving in weight of 1150 lb.

This bus, with the hydraulic drive, was placed in regular service on Aug. 18, 1933, and has been in regular operation since then. During the greater part of this period, the bus has been operated over a very heavy, frequent-stop city-run. Up to Dec. 28, 1933, it had been operated a total of 17,577 miles with the converter. The average gasoline mileage of the converter-equipped bus is tabulated below in comparison with corresponding average gasoline mileage obtained with the gasoline-electric buses of the same type as the one converted to hydraulic drive:

Average Gasoline Mileage in Miles Per Gallon

1933	Hydraulic Bus	Fleet Average of Gasoline-electric Buses	Best Individual Gasoline-electric Bus
August	5.00	3.91	4.11
September	4.87	3.82	3.89
October	4.60	3.71	3.79
November	4.36	3.50	3.66

All of the work in connection with this installation has been under the direct supervision of W. W. Kunz, master mechanic of the International Railway Co. According to Mr. Kunz, no difficulty has been experienced with the converter in the present installation other than a slight bit of trouble with the free-wheel clutch when cold weather set in, due to the use of too heavy a grade of oil. This trouble was immediately corrected by changing to a lighter oil.

The equipment available at the shops of the International Railway Co. has enabled Mr. Kunz to make block tests of the converter. From his test and operating experience to date he has reached the conclusion quoted below from a recent statement made by him in response to a request for information concerning the performance of the converter-equipped bus:

"The ability of the hydraulic-drive bus to make a given scheduled speed is better than that of a gasoline-electric due to about 25 per cent greater efficiency in power transmission. Tests which we have conducted on the latest types of mechanical-drive buses indicate that, for equivalent bus weight and engine horsepower, the hydraulic drive, like the gasoline-electric, will maintain considerably faster scheduled speeds due to the simplicity of driving and the very high initial acceleration rate."

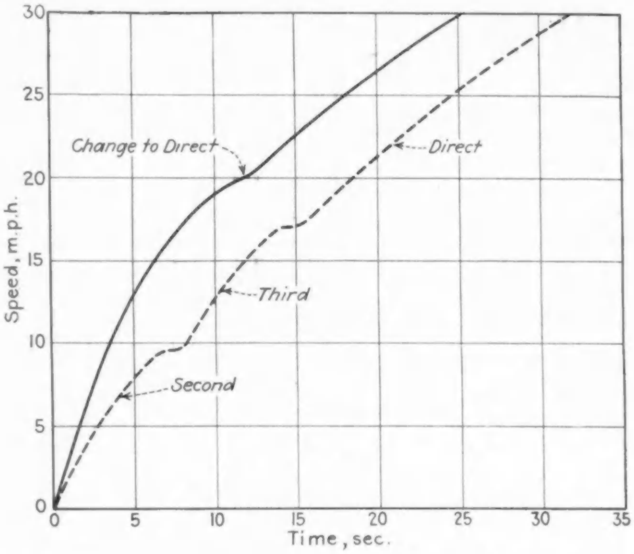


Fig. 9—Comparative Acceleration Curves Given by the Lysholm-Smith Converter and Gearbox

The full-line curve is the acceleration curve of a fully laden torque-converter-equipped Titan bus. The smooth acceleration should be noted. The dash-line curve results from data taken on a lightly laden Titan bus equipped with a gearbox, and is a curve which rarely could be improved without considerable discomfort to passengers

In making the Buffalo installation, Mr. Kunz has applied vacuum servo-motor control to the clutch-shifting mechanism for changing from hydraulic drive to direct drive and vice versa, and has arranged the control so that the converter is normally in hydraulic drive and shifts to and operates in direct drive only when and so long as the vacuum control-pedal, which is a small pedal similar to an accelerator pedal and placed in the position of the usual clutch pedal, is depressed by the bus operator. This arrangement has been found to be exceedingly satisfactory from the standpoint of the driver when operating in frequent-stop service, and Mr. Kunz states that his drivers prefer the hydraulic-equipped bus to either the gasoline-electric buses or the gear-transmission buses.

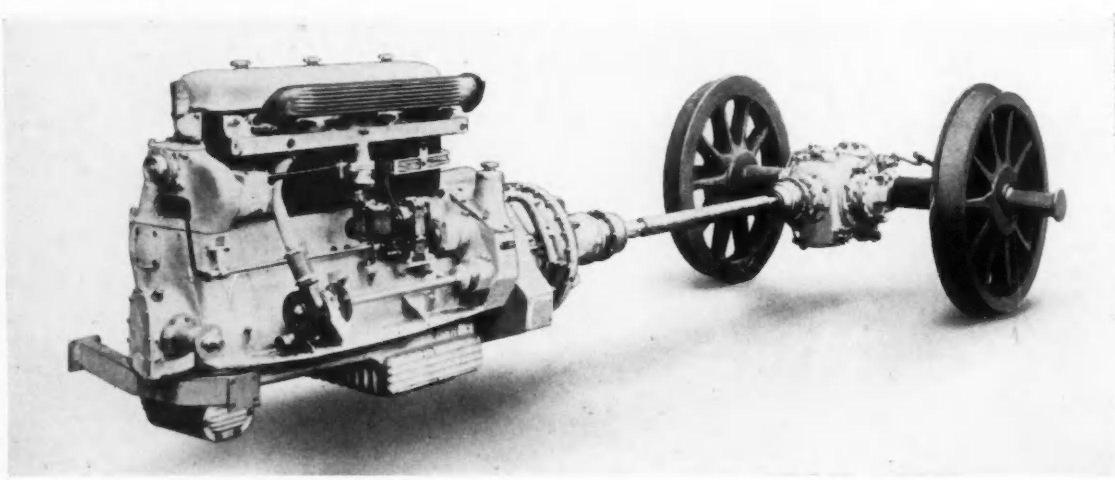


Fig. 10—A Lysholm-Smith Converter as Built by Leyland Motors, Ltd., for Railcar Drive